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ARTICLES on design, construction and operation of oil-engines and motorships by the world's foremost writers on marine engineering.

Motorship

ILLUSTRATIONS of the newest designs in international merchant motorship and Diesel-engine construction and auxiliary equipment.

Vol. XI April, 1926 No. 4

Big American Engine Ends Hard Test

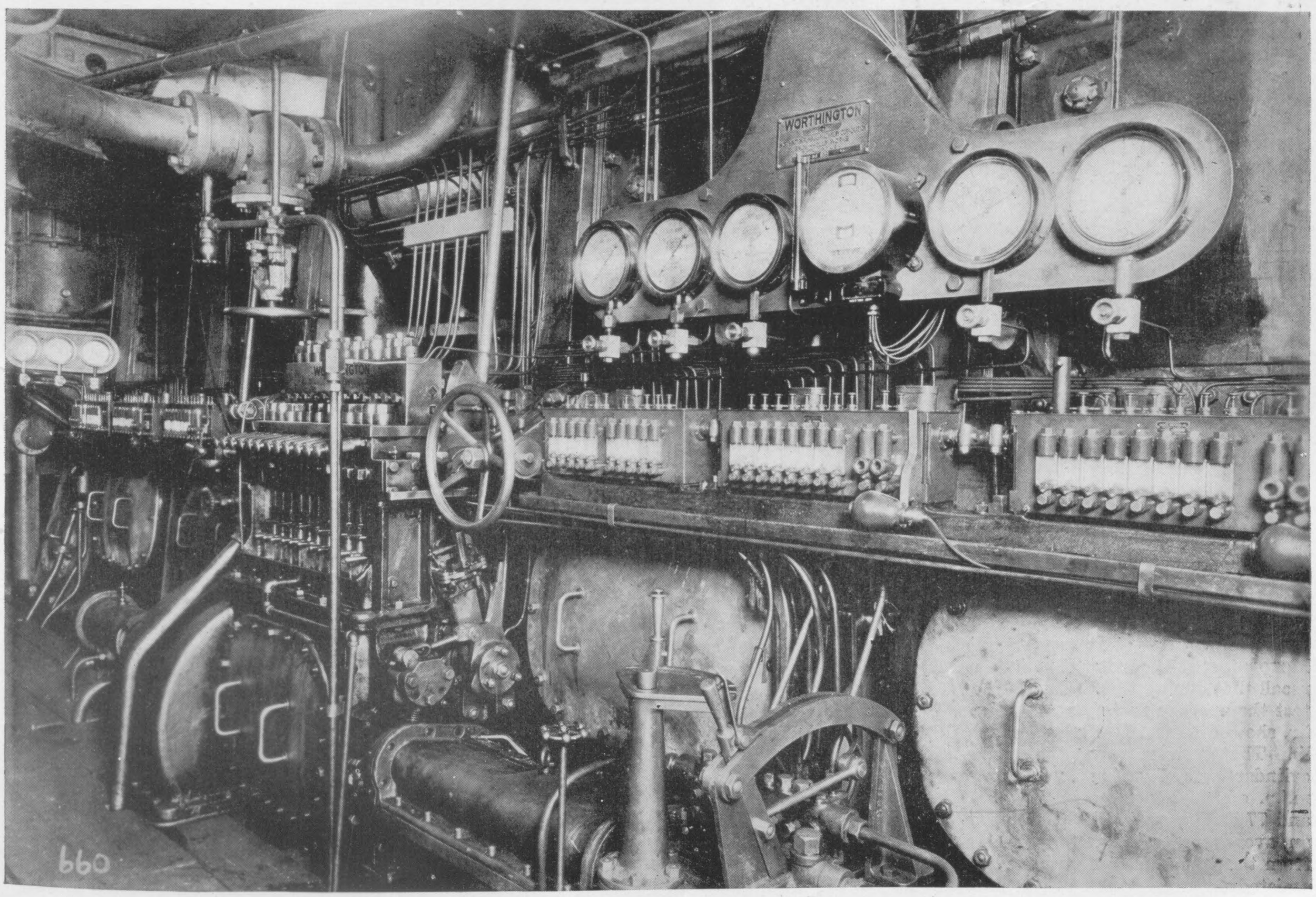
Double-acting Engine of 2900 s.hp. Designed and Developed by Worthington Concludes 30-Days' Non-Stop Run Under Shipping Board Supervision

CROWNING five years of persistent development, the conclusion of the strenuous trials of the first 2900 s.hp. Worthington engine for the Shipping Board has definitely set the seal of success on the double-acting design evolved by the engineers of the Worthington Pump & Machinery Corporation as an all-American production.

On March 4 this engine completed a 30-days' full power endurance test followed by rigid maneuvering trials under the supervision of the Shipping Board's inspectors. Contract power was maintained throughout the month-long trial; lubricating oil consumption was well under the guarantee, but fuel consumption a trifle higher; and operation was up to the best standards of smoothness and quietness. The trial data will not be released by the Shipping Board until the dynamo used for the tests has been recalibrated after the trial of the second Worthington double-acting engine which is due to undergo test before long.

With the general features of the Worthington double-acting engine readers of this magazine are already acquainted. They were first disclosed in MOTORSHIP of Sept., 1924. The subsequent publication of the excellent exposition of the development of large oil engines (MOTORSHIP, Jan., Feb., March, April and May, 1925) by Dr. Charles Edward Lucke, Professor of Mechanical Engineering at Columbia University and consulting engineer to the Worthington firm, elucidated the intrinsic importance of the double-acting principle. And the review in the Jan., 1926 MOTORSHIP of the paper by O. E. Jorgensen, consulting engineer to the same company, indicated to readers where a further consideration of the problem could be studied.

As far back as 1921, work was commenced on an experimental horizontal Diesel engine having a single cylinder 14 in. diameter by 19 in. stroke. This was designed for operation at 150 r.p.m., but after



Control stand of the 2900 s.hp. Worthington 2-cycle double-acting engine that has just passed a 30-days' continuous full power trial

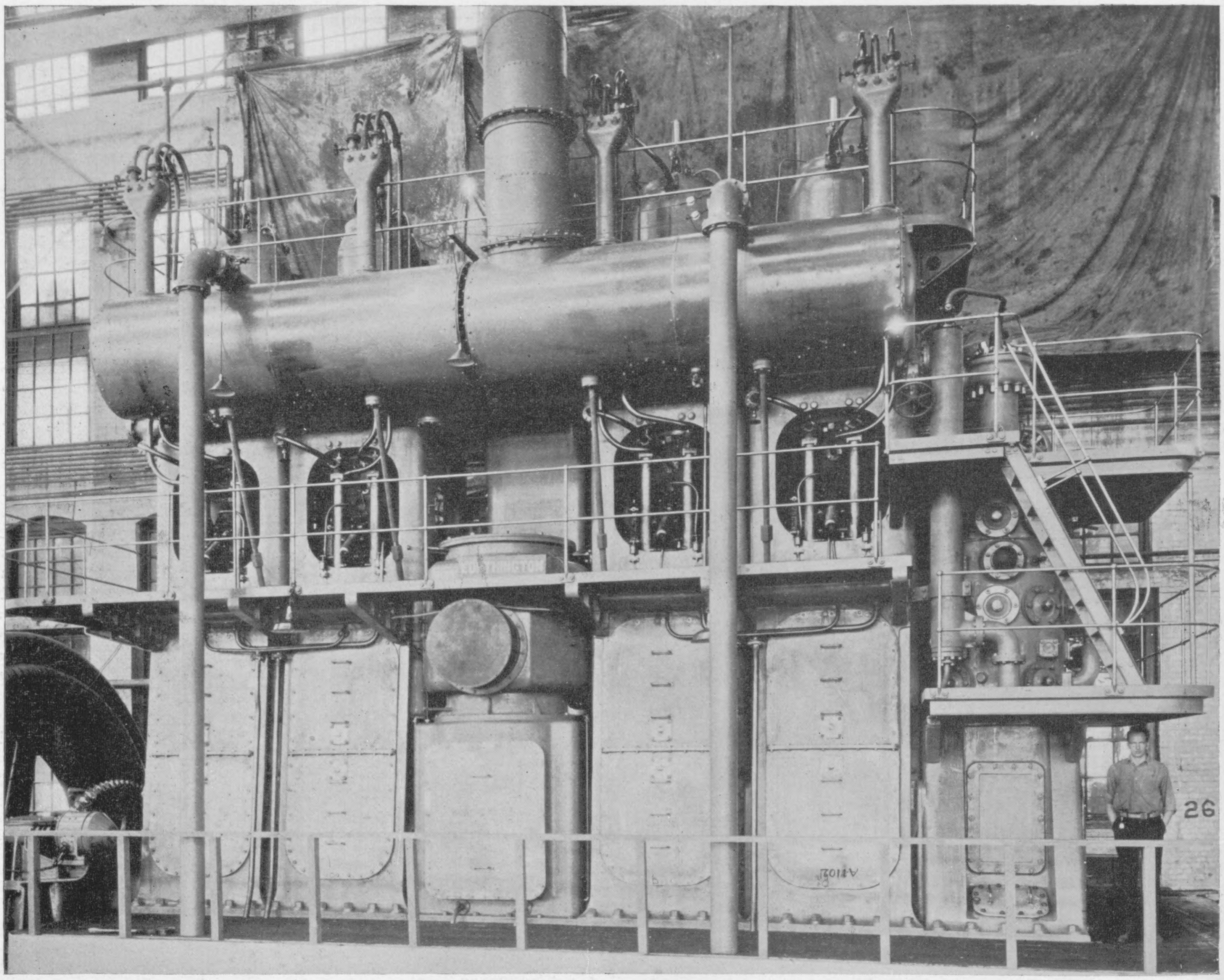
some initial tests the speed was raised to 200 r.p.m. in order that the processes of combustion and scavenging, as well as their effects on the combustion space, piston rod packing, etc., at all piston speeds approximating those of larger engines, could be studied.

Data obtained from the horizontal engine were utilized in the design and construction of a vertical engine of marine type with a single cylinder 27 in. diameter by 40 in. stroke. This engine was run at full load within one month of its initial start in June, 1924, and three months later

The endurance run of this latest and successful commercial engine really started on Feb. 2, the engine being operated at about full power for a couple of days. At the end of that period the official trial started, requiring the engine to develop continuously 2900 s.hp. at 95 r.p.m. from Feb. 4 until March 4, when without interruption, the load was increased 10 per cent for a period of 6 hours and then for a further period of 4 hours the speed was increased 5 per cent.

Following the long endurance run, maneuvering trials were carried out. Com-

a speed of 95 r.p.m. The center block of each cylinder is supported on a distance piece or box resting on independent cast-iron box frames, which in turn rest on the engine bedplate. Each frame is flanged to the adjoining frame to form a common crankcase for the power cylinders and compressors, but the frames are only tied in on the bottom, permitting easy expansion of each vertical element. The cylinders are held by means of four tie-rods placed at each corner of the cylinder center block and passing through the distance piece, the frame members and the bedplate. The



Back of the big Worthington double-acting engine on the test floor of Buffalo preparatory to the long full power trial just completed

it made a 30-day non-stop run under the supervision of U. S. Shipping Board engineers.

Results of this test were published in MOTORSHIP at that time, but it is useful to recall that during the 30 days' (720 hours') test the average fuel consumption per i.hp.-hr. showed the relatively low figure of 0.339 lb. The average power from the single cylinder was 778 i.hp. with an average m.i.p. of 81 lb. per sq. in. for the top end and 77 lb. per sq. in. for the bottom end at an average of 89.8 r.p.m. Data obtained from the large single-cylinder vertical unit assisted very materially in the development of the 4-cylinder engine, which has just completed its trials.

plete reversal from full load in one direction to full load in the opposite direction was carried out in 6 seconds. A total of 28 consecutive reversals were made in an average time of $8\frac{1}{2}$ sec. each, and 14 reversals were made in one hour. A maximum of 47 reversals was made without drawing upon any air other than the supply in the maneuvering air tank. Asphalt-base bunker oil was used, requiring heating before it would flow through the piping. After having been shut down for 17 hours the engine was started up instantly without the fuel valves being primed.

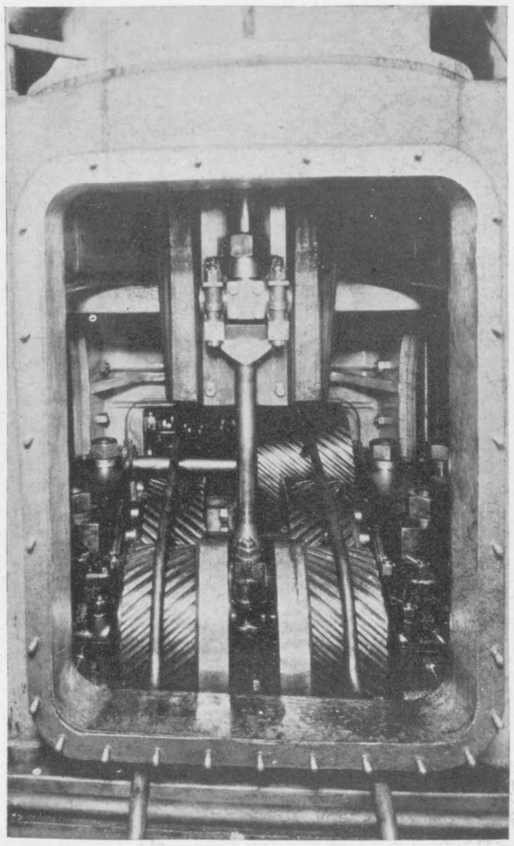
The engines for the Shipping Board have four cylinders 28 in. diameter by 40 in. stroke, with a rated output of 2900 s.hp. at

rods take up the tension stress resulting from combustion in the upper ends of the cylinders, and the compressive stresses in the frame caused by the lower cylinder ends are taken up by the cast-iron members. The frames carry the support for the crosshead guides, which are of the flat water-cooled type with astern bars to support the crosshead shoes when the engines are running astern.

The crankshaft is built up of sections drilled to provide the supply of lubrication to the journals, crankpins, and crosshead pins under pressure. Forced lubrication is also applied to the crosshead shoes and to the piston-rod stuffing box. The bearings are babbitted steel shells and are removed

by rolling after being relieved of the weight of the shaft.

The cylinder comprises two dome-shaped working portions *AA*, with a common central block *D* between them containing the exhaust and scavenge air ports. The dome shape of the cylinder ends is adapted to withstand internal pressure, and their extensions enshroud cast-iron liners that take the wear occasioned by the movement of the piston, but are not called on to withstand any radial pressure stresses, all of which are carried by the surrounding steel shell. The inner ends of both upper and lower cylinder liners *CC* are thickened in way of the exhaust and scavenge ports, and the block *D* has an internal ring-shaped ledge upon which the machined face of the inner ends of the liners rest. Liners, steel cylinders and block are all clamped into a single assembly by lugs *E* and studs *F*, the lugs bearing against the formed inner ends



Gear drive for scavenging pumps

of the steel cylinder. This construction permits of expansion of liners and shell without disturbing the central assembly. A light case or cover *H* is slipped over the cylinder and bolted at the outer end to the flange of the cylinder cover, while the inner end is packed against leakage by a light gland and fibrous packing ring. This forms the water cooling jacket.

The jacket surface of the steel cylinder is corrugated to intensify cooling, which is further increased by the introduction of water through the central block. Flowing through the passages in the block the water enters drilled holes in the inner ends of the liners. Retarders consisting of steel rods in the holes increase the water velocity to about 3 ft. per sec., resulting in a high rate of heat transfer from the liner and steel cylinder. After passing through these holes, the water flows along the space between the jacket cover and the steel cylinder, the amount of water being regulated by valves at the outlets from the cylinder ends.

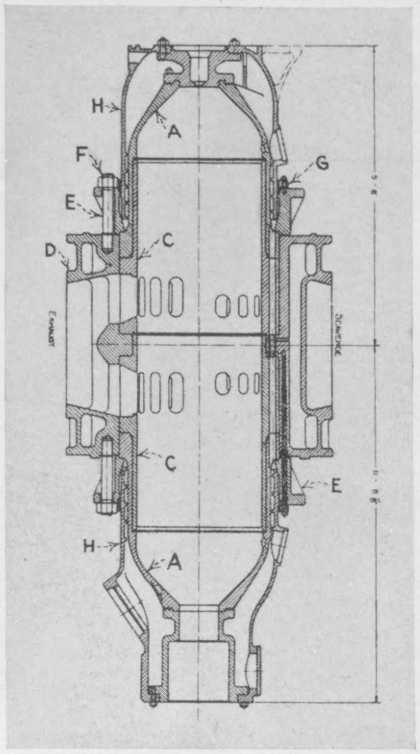
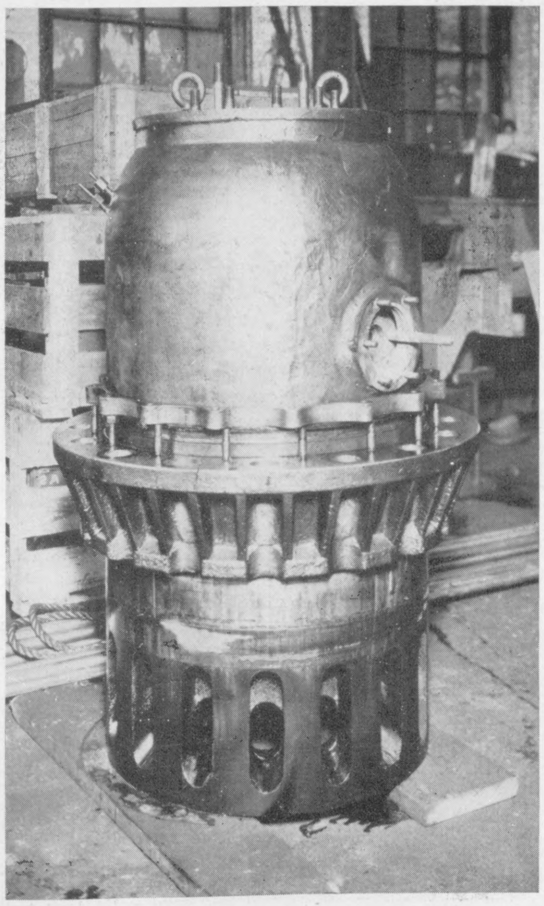


Diagram of Worthington cylinder construction

Pyrometer readings taken during the 30-day test on the single cylinder engine go to prove that the cooling system does all that is claimed for it. One of the illustrations accompanying this article shows the location of the pyrometer elements and their temperature readings during test. Attention is called especially to the readings taken on the jacket side of the combustion chamber at a point about 0.01 in. beneath the inner exposed surface. The temperature gradient through the metal is low, making internal heat stresses light, and this is traceable to the effective cooling water system and to the thin section made pos-

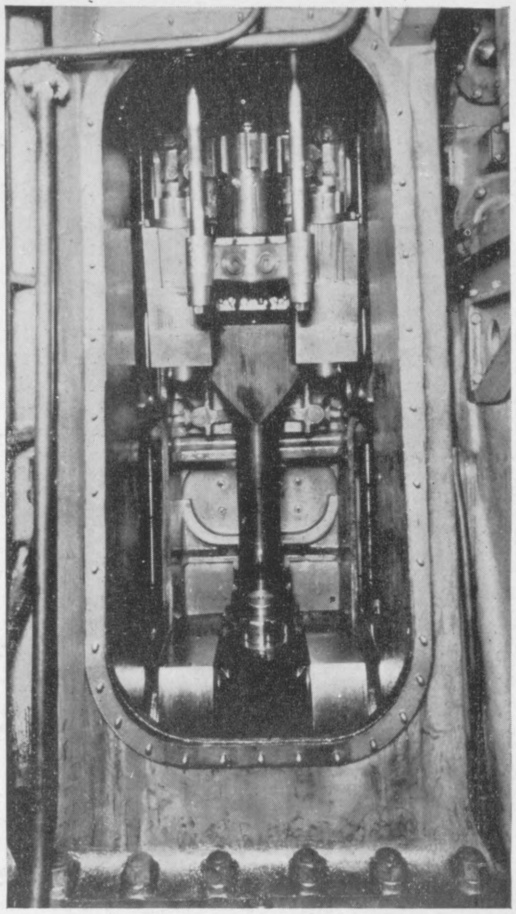


Upper part of the cylinder assembly

sible by the use of the steel cylinder shell.

The piston is composed of five parts. The two end sections exposed to the heat of combustion are of alloy steel, and between them are placed two cast-iron bodies which hold the piston rings. This makes four parts fastened to the piston rod, and in the space between the two assemblies are clamped the two halves of a cylindrical central section held together by keyed bolts. This permits the parts to be taken adrift very quickly for inspection.

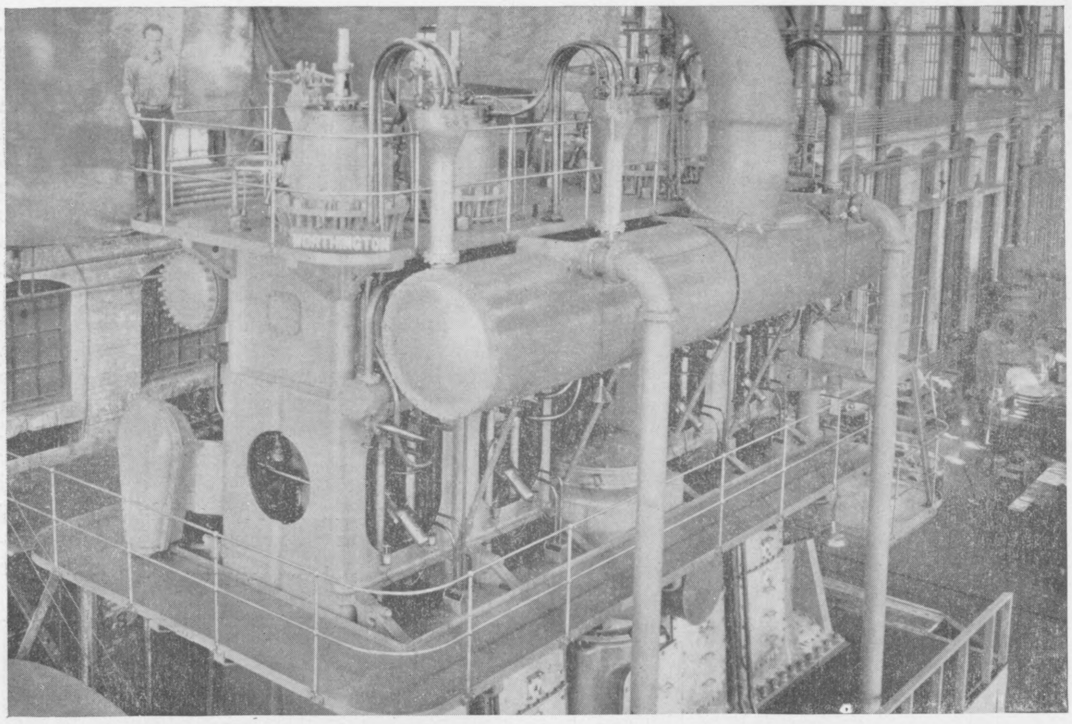
The piston rod is hollow, provided with a central tube to convey cooling water to the piston head, the return being made through the space between the tube and the wall of the rod. The water reaches the rod by means of telescopic tubes leading to passages in the crosshead. The stuffing boxes of these tubes are arranged outside the crankcase enclosure, thus eliminating the danger



Crosshead with piston cooling connections

of cooling water mixing with lubricating oil in the crankpits.

Particular success has been obtained in this engine with the elimination of leakage along the piston rod, and the design of stuffing box and packing rings adopted has led to no trouble. The rings are cooled both by heat transfer to the water flowing through the piston rod and by a liberal application of lubricating oil, this latter being recovered. Metallic packing rings have been brought to a high state of development during their use for some 30 years in large double-acting blast-furnace gas engines, and for nearly 20 years in stationary double-acting Diesel engines of the 4-cycle type. The piston rod in the largest blast furnace gas engine built by Worthington is 15 in. diameter, whereas in the new Worthington Diesel engine the rod is approximately 9 in. diameter. Early criticisms that the double-acting Diesel engine



Middle grating and upper grating of 2900 s.hp. Worthington double-acting engine

would always experience trouble at the point where the piston rod passes through the bottom cylinder cover sprang from the ignorance of steam engineers. No problem ever was involved in that.

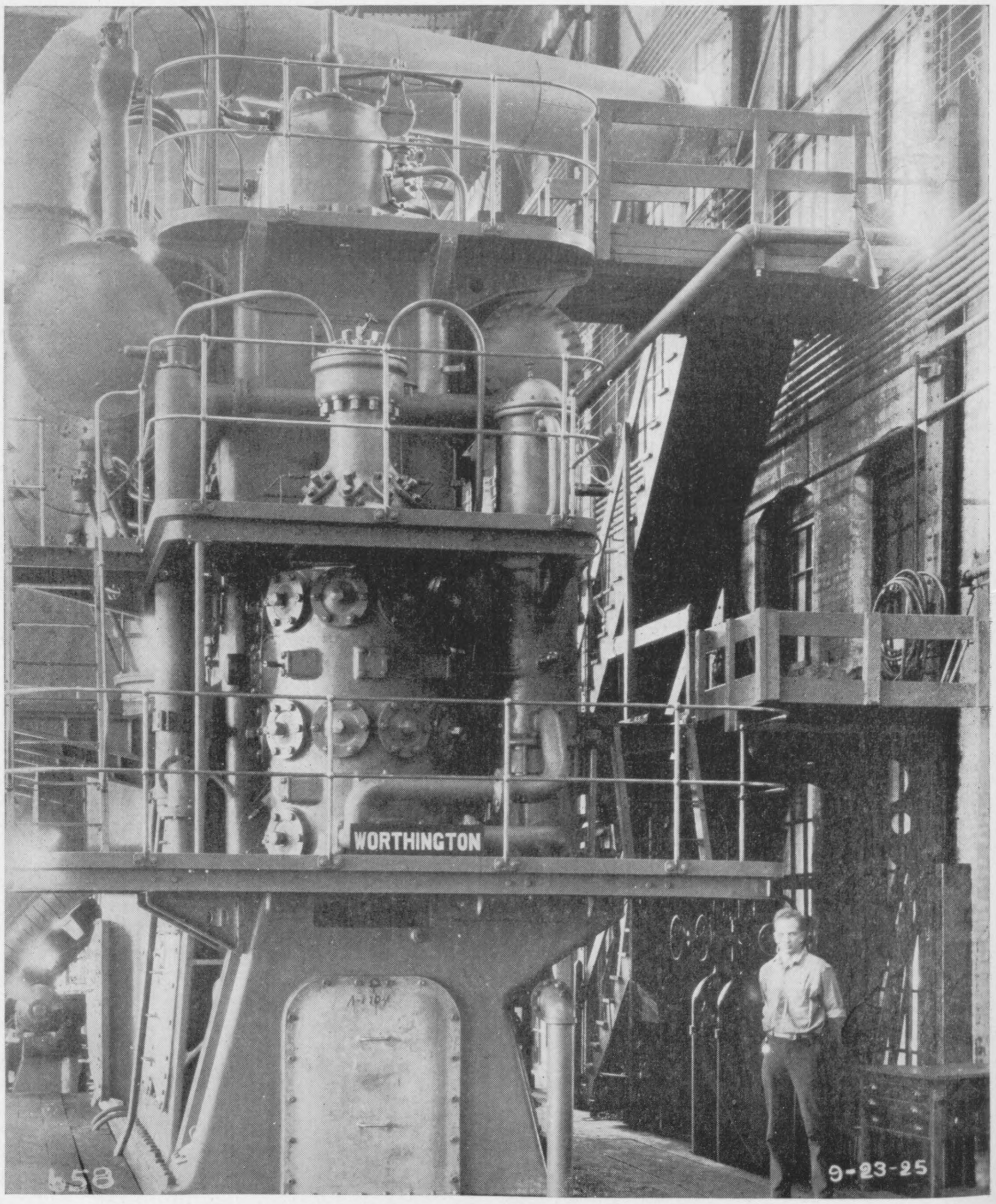
The fuel valves are of the needle valve type with atomizing discs. Oil is introduced above the discs by a governor controlled pump, and when the needle valve is raised by the cam and rocker arm, injection air forces the fuel into the cylinder in the form of a spray. The top fuel valve is arranged along the cylinder axis, resting in a forged cover plate which seals the opening in the top cylinder end and to which also is bolted the light outer cooling jacket cover. The only part of the fuel valve exposed to the heat of combustion is the comparatively small tip, the rest being surrounded by the water cooled cover. For the lower combustion space fuel is supplied by two valves since, owing to the piston rod, one cannot place a spray valve in the center. These two valves are in an inclined position. Each valve tip has two holes so placed that fuel does not strike the piston rod itself, but passes on either side of it, there being thus four fuel sprays. The valve bodies in the bottom combustion space are equipped with a sleeve into which the oil is introduced tangentially, and the resultant turbulence maintains the presence of oil around the tips. This overcomes the tendency of the oil to flow downward and away from the tip. The fuel valves for the bottom combustion space are actuated through rockers and push rods by cams placed on a camshaft running along the engine above the frame. This camshaft also operates the push rods of the rocker arms of the fuel valves for the top combustion space.

Reversal of the valve timing gear is carried out in the Worthington engine by displacement of the helical gears between the crankshaft and the camshaft. The fuel pump is also driven by the same helical gear train and the timing of the pump plungers therefore altered to conform with the desired direction of running. The fuel pump consists of a steel body provided with a plunger for each engine cylinder end. The amount of fuel delivered is controlled by lever action which regulates the timing of the pump suction valve while an overspeed governor is also part of this mechanism.

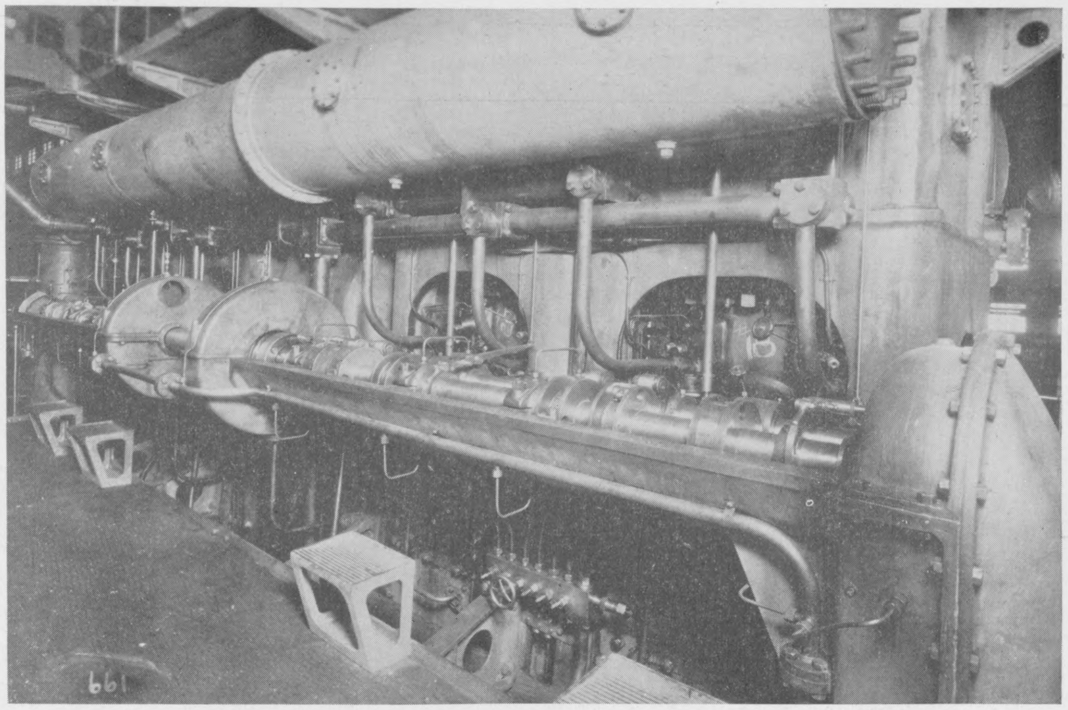
The scavenge pumps are two double-acting cylinders placed in tandem and arranged between each pair of cylinders. The designers selected the reciprocating type of pump in place of the turbo blower for this engine, because of its small power requirements, although the latter would reduce the overall length of the engine. The arrangement adopted reduces the pump diameter while avoiding a long stroke, and the pump is driven at twice the engine speed. The stroke is only 20 in. which results in the same piston speed as the power cylinders. The bulk and weight of the auxiliary are thereby diminished. The pump is designed to deal with 15,000 cu. ft. of free air per min.

Mechanically operated valves of rotary type are used, working under a low pressure and easily lubricated. The pump pistons are made of an aluminum alloy, without rings, air leakage being negligible by reason of the slight pressure difference.

The injection air compressor is of the 4-stage type, of which the first is double-acting, and each stage discharges its air into a straight tube cooler. It is calculated that the fourth stage reduces the temperature increase between the initial intake and



Compressor end of the 2900 s.hp. Worthington engine for the Shipping Board



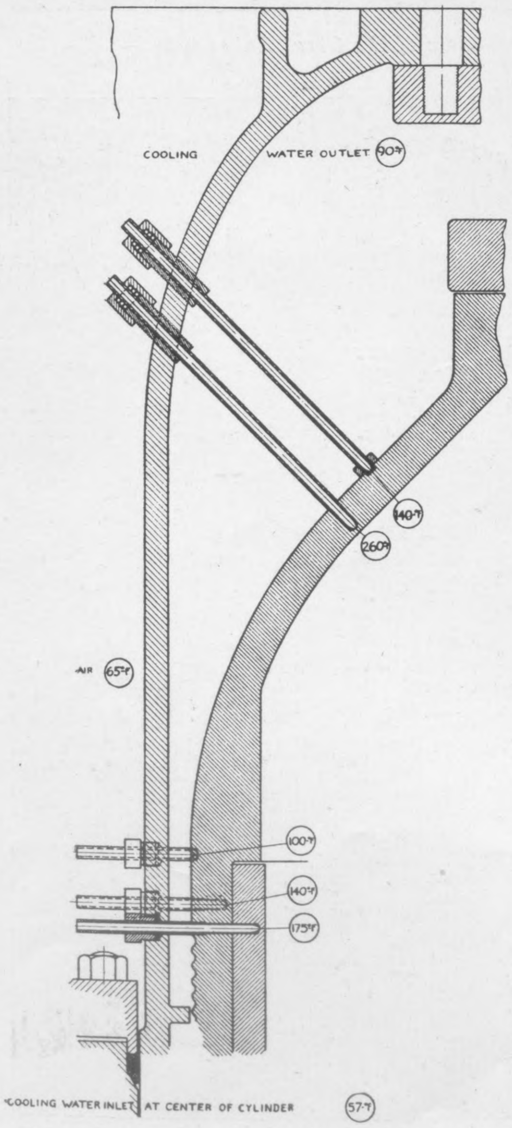
Camshaft at the level of the lower cylinder covers of the engine

the discharge into the after cooler by 23 per cent, and the power consumption by 20 b.hp. in comparison with a 3-stage compressor of the same capacity.

A regular program of double-acting Diesel engine construction is being adopted by the Worthington firm. There is a series of engines already in hand with cylinders of 16 in. diameter and with 24 in. stroke, rated at 250 b.hp. per cylinder at 180 r.p.m. Thus a 4-cylinder unit of this latter type will give about 1000 b.hp. A 3-cylinder engine of this type to give 750 b.hp. has been ordered for one of the U. S. Steel Products Co. boats now building in the Federal yard at Kearny, N. J.

In his address before the Society of Naval Architects and Marine Engineers in New York last November, O. E. Jorgensen summarized his view of the potential superiority and future applications of the 2-cycle double-acting marine engine in the following statements:

"If 2-cycle double-acting engines were simply substituted for 4-cycle single-acting engines in motorships, the earning capacity of such ships would be increased due to the lighter machinery, and operating expenses would be reduced due to the lower first cost reflected in lower insurance, depreciation and interest charges, to the smaller amount of machinery to be cared for and kept in repair and to the elimination of a special crew for exhaust valve grinding as used in 4-cycle ships. But such a substitution disregards entirely the very considerable reduction in space requirements of this engine compared with single-acting types, and it is natural to expect that 2-cycle double-acting motorships will take advantage of the possibility of a return to single-screw installations instead of using the twin-screw arrangements to which 4-cycle single-acting ships have been forced by bulky engines.



Test elements and temperatures in cylinder

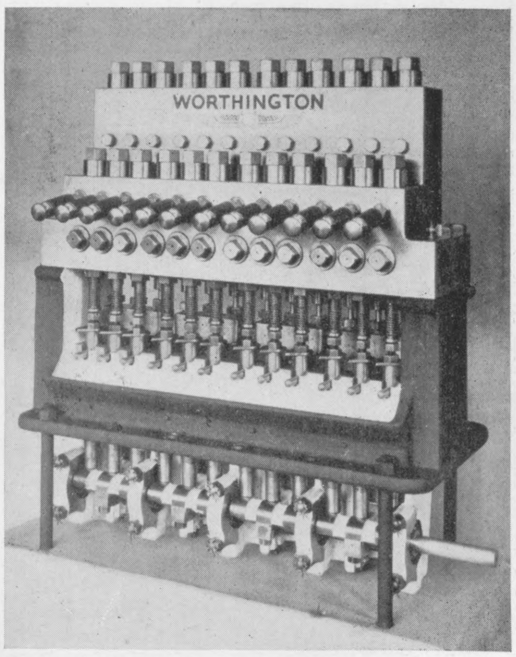
"This would mean a further simplification, which should be reflected in the crew wages and repair cost. For the hull it means a reduction in cost due to the simpler stern, the single propeller, stern tube, shaft, shaft alley and engine foundation. With all the assertions to the contrary, it is still the general belief that the propeller efficiency of single-screw ships at cargo ship speeds is better than that of twin-screw installations, and it should not be overlooked that the steamer has continued to adhere to the single-screw system despite the twin-screw example given by some of the present motorships. Important as this is in the case of new ships, it is even more so in the case of conversions of steamers to motor drive, because such steamers without exception are single-screw vessels, in which a twin-screw installation is not feasible for economic reasons. Conversion of steamers is an important field in this country, due to the existence of the numerous Shipping Board vessels in need of more efficient engines, but it should be expected that conversions will come more and more to the forefront in all countries with the reduction in conversion cost due to the 2-cycle double-acting engine and to the more general use and proven reliability of motorships. Until quite recently the field of fast passenger liners was reserved for steam power, but even here the Diesel engine is now making inroads. Several high-powered motorships are now building abroad (two are now in operation), and their performance in service will be followed with keen interest by all marine engineers.

"The most powerful engine to go into any of these ships has eight double-acting 4-cycle cylinders 33 in. by 59 in. It is to work at 115 r.p.m., 1130 ft. per min. piston speed, and to develop 10,000 i.h.p., or 7500 b.h.p.

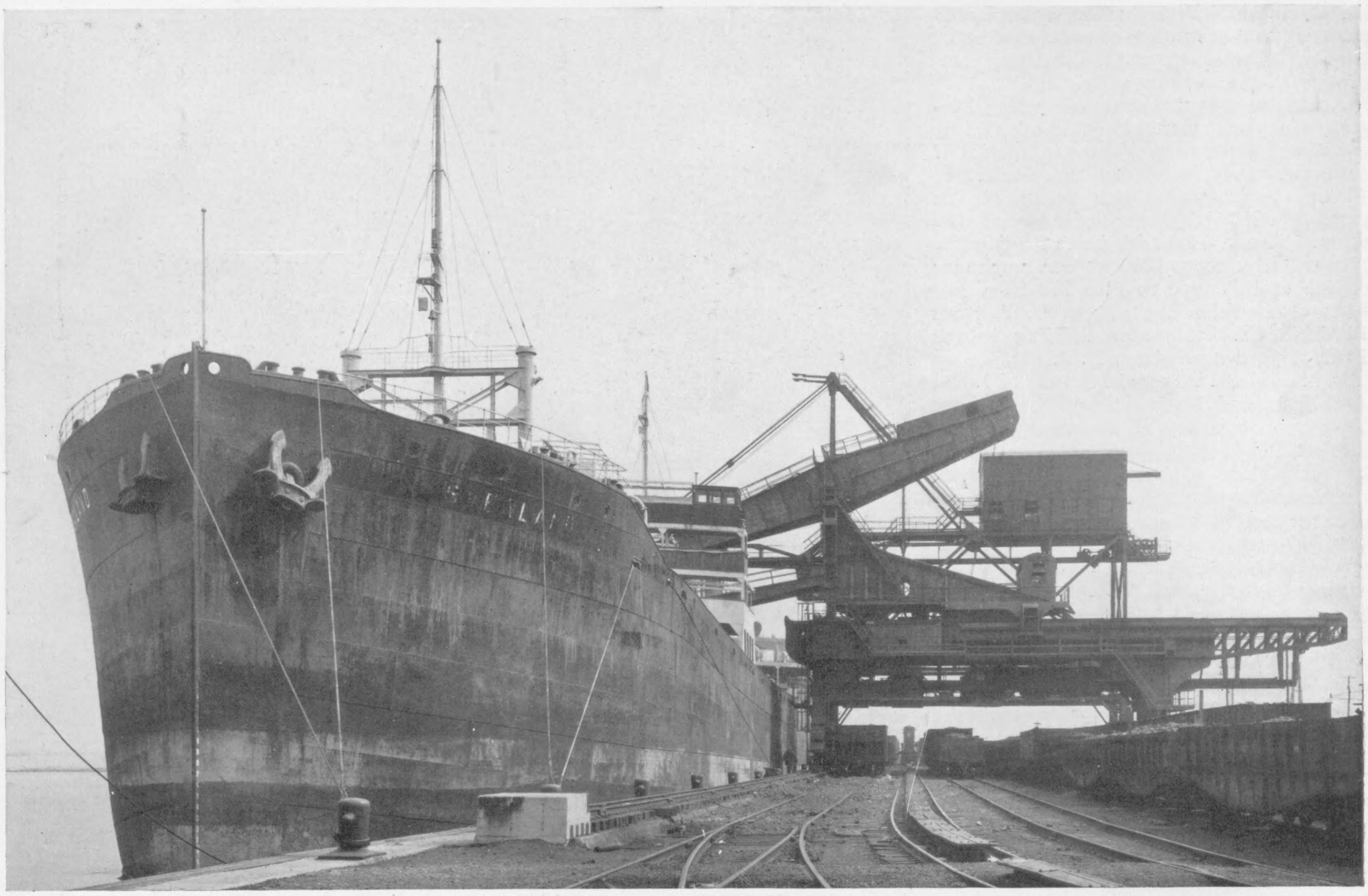
"The Worthington 2-cycle double-acting engines now building for the U. S. Shipping Board, are able to develop the same power for continuous operation per cylinder, including its compressors, when working at 125 r.p.m., or 830 ft. per min. piston speed.

"The above-mentioned 4-cycle engine represents probably the maximum dimensions of which a reciprocating engine can be made and cared for in a ship, and exhausts the possibilities of the 4-cycle double-acting system. The engine described above has still a comfortable margin in power, and a 34 in. by 48 in. cylinder to work at 110 r.p.m. and to develop 1500 b.h.p per cylinder is proposed as the present maximum of this type.

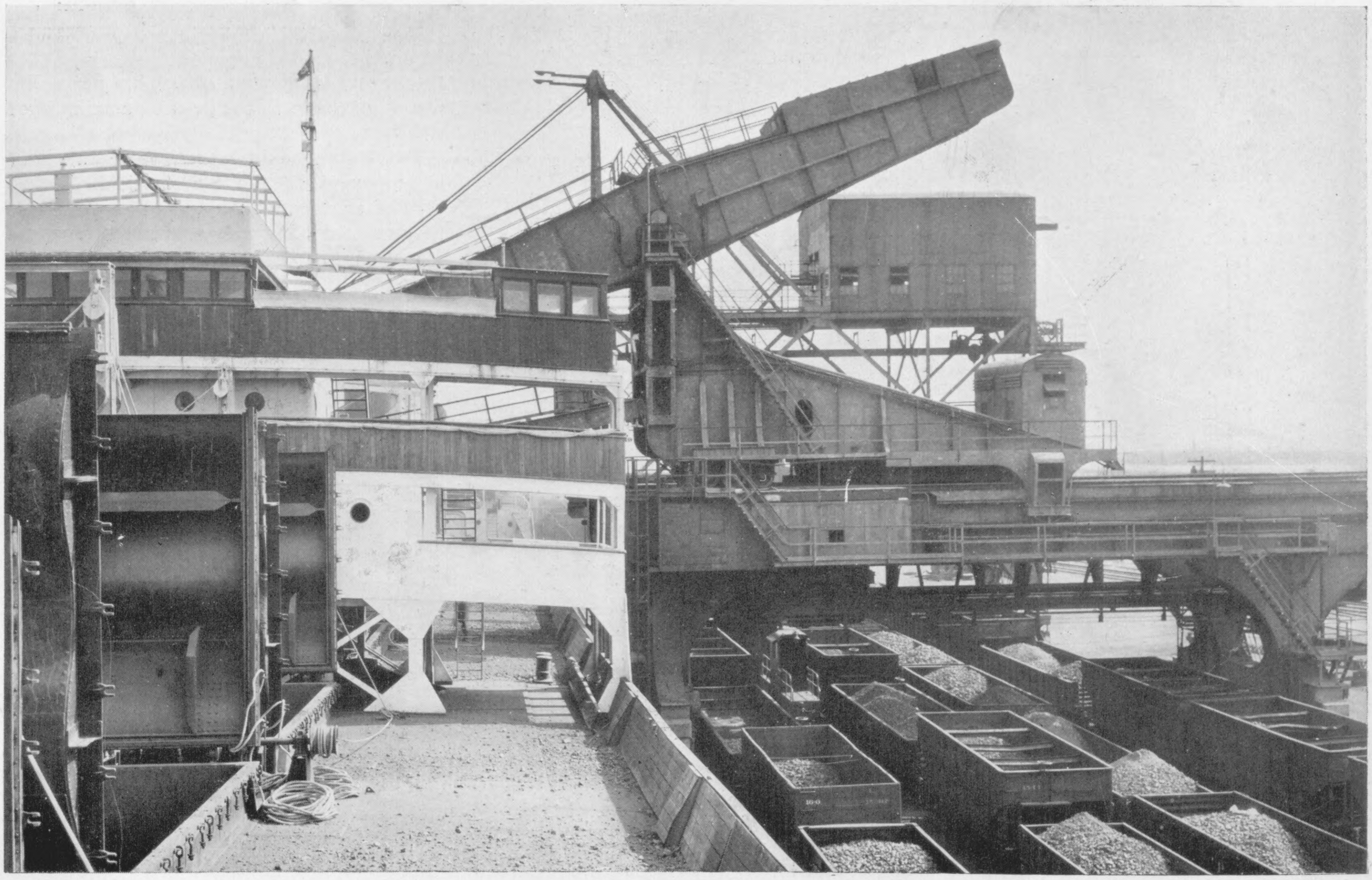
These statements express the opinions of a proponent of the 2-cycle double-acting type of engine and are quoted here because they present admirably the thoughts of the school of progress to which he belongs, but they will be understood to be controversial.



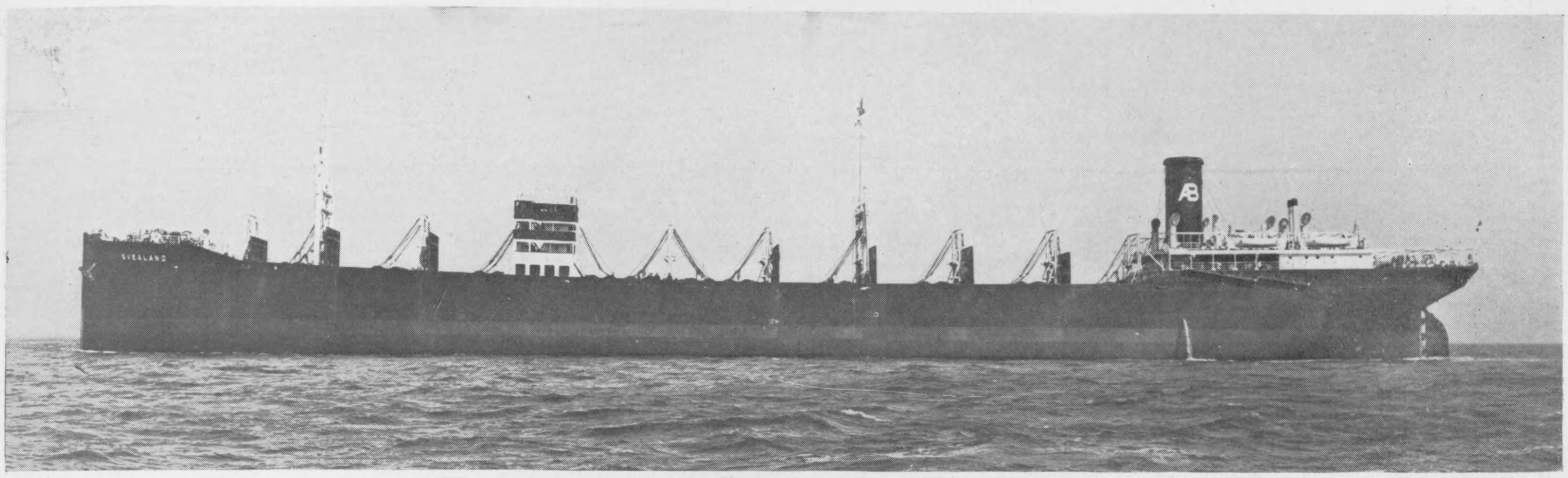
Fuel pumps for the 12 fuel valves



M.s. Svealand alongside the pier at Claremont, N. J., discharging ore through rapid unloaders of the Great Lakes type



Direct from the holds of the Svealand into the railroad cars for the short haul to the furnaces of the Bethlehem Steel Co.



Picture of the m.s. Svealand in ballast, showing some of her hatch covers raised

Motor Freighters of 21,000 Tons D. W.

Broström Ships Svealand and Amerikaland Running Regularly Between North Atlantic Range and Chile

WITH the operation of the motor-vessels SVEALAND and AMERIKALAND the biggest freight steamers in the world are now matched by motor freighters of the same huge size. These big vessels have a deadweight capacity of 21,000 tons and belong to the same class as the ore carriers MARORE, BETHORE, CHILORE, etc., operated by the Ore S. S. Co. between Chile and this country via the Panama Canal.

SVEALAND and AMERIKALAND are not under the American flag, being owned by Axel Broström & Son of Gothenburg, Sweden, but they are under charter for 20 years to carry iron ore from Cruz Grande, Chile, to Sparrows Point, Md., and New York for the Bethlehem Steel Co. That they are proving more economical in operation than the big ore steamers in the same trade is not to be doubted, but because the ore steamers represent a big investment and at the present time are not suitable for any other trade, it is pretty sure that the steamers will continue to operate for many years, notwithstanding the economy of the motorships.

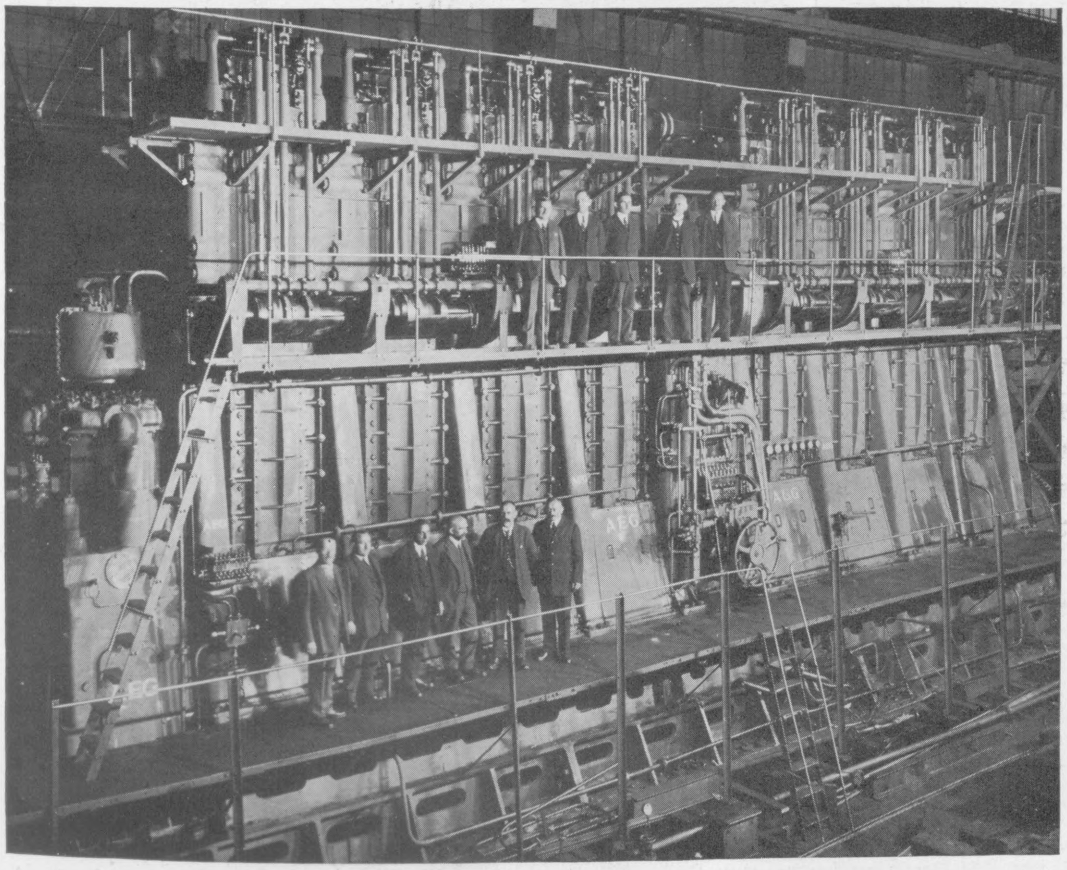
Ownership of these vessels is one more instance of the courage of the late Dan Broström. In addition to the tonnage operated under the firm name, Axel Broström & Son, which includes two other motorships besides the SVEALAND and AMERIKALAND, Dan Broström's motorship interests extended to the Swedish East Asiatic Company under the flag of which he had three motor vessels, the Swedish Orient Line with one motorship, the Swedish American Mexico Line with two motorships and the Swedish American Line with one motorship and with the big motorliner GRIPSHOLM in service. There are also three motorships building for these various companies. He was one of the first shipowners to demonstrate his confidence in the Diesel engine and remained a leader in motorship operation to the end. That he should have ordered two motorships equal in size to the biggest freight steamers in operation was just one more development of his advanced policies.

In the design and construction of the SVEALAND and AMERIKALAND the foreign builders had the benefit of all the experience which the Bethlehem Shipbuilding Co. had gained in building the big American steamers and which the Ore S. S. Co. had obtained in operating them. This was freely placed at the disposal of the Broström firm and used by the Deutsche Werft in Hamburg, the builders of the two motorvessels. The American steamers and the Swedish motorvessels are all of a size, with possible minor variations. The leading characteristics are as follows:

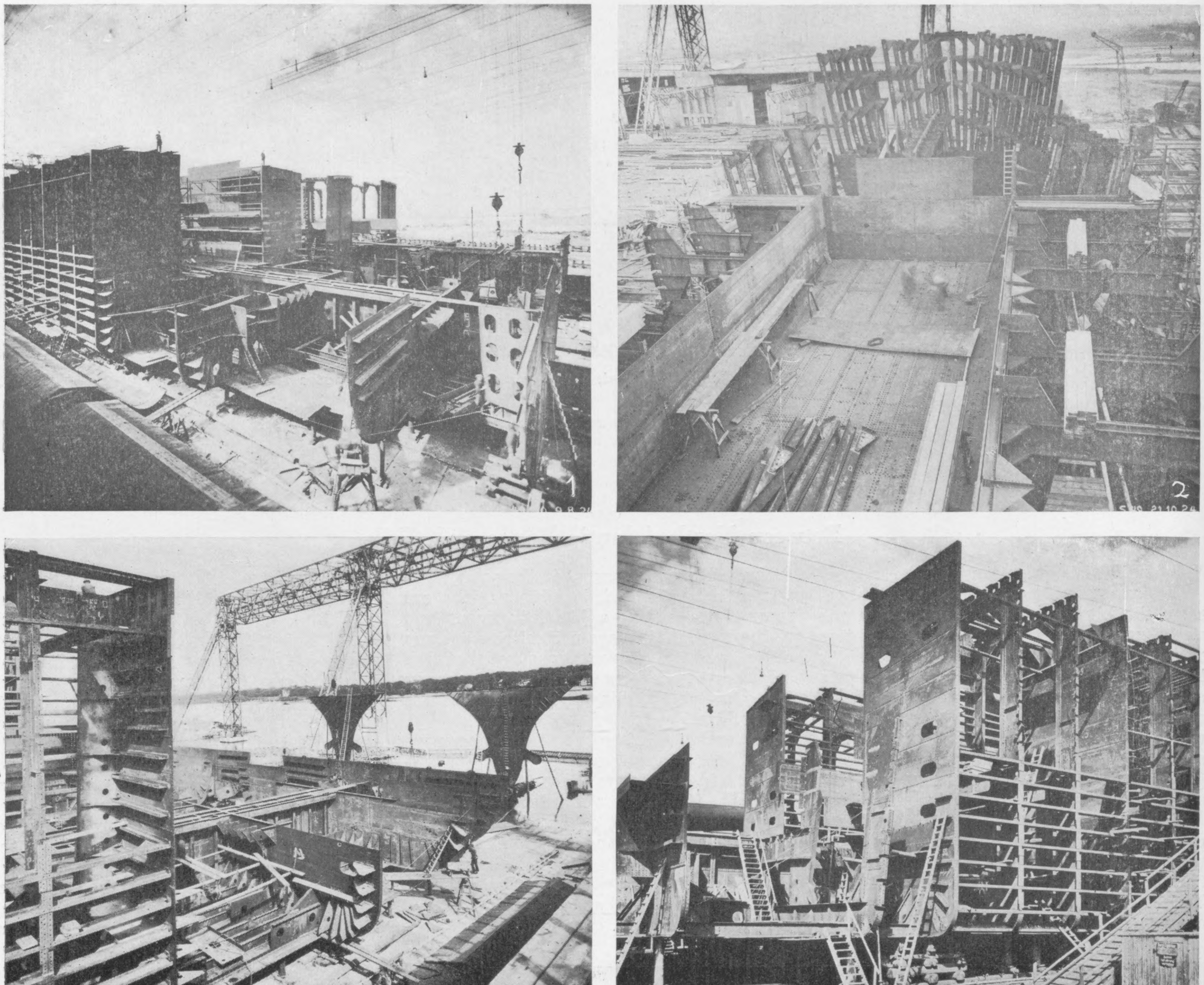
Svealand and Amerikaland

Length, overall	571 ft. 7 in.
Length b.p.	548 ft. 1 in.
Molded breadth	72 ft. 0 in.
Molded depth	44 ft. 0 in.
Extreme draft	32 ft. 3 in.
Displacement	30,000 tons
Weight of ship	9,135 tons
Deadweight capacity	20,865 tons
Height of double bottom	14 ft. 0 in.
Gross register (Swedish)	15,339 tons
Net register (Swedish)	4,377 tons
Net register (Panama Canal)	3,826 tons
Length of ore holds	367 ft. 0 in.
Depth of ore holds	24 ft. 2 in.
Width of ore holds	30 ft. 9 in.
Total power	4,800 s.hp.
Type of engine	4 cycle 8-cylinder
Cylinder diameter	29.13 in.
Piston stroke	47.24 in.
Engine speed	110 r.p.m.

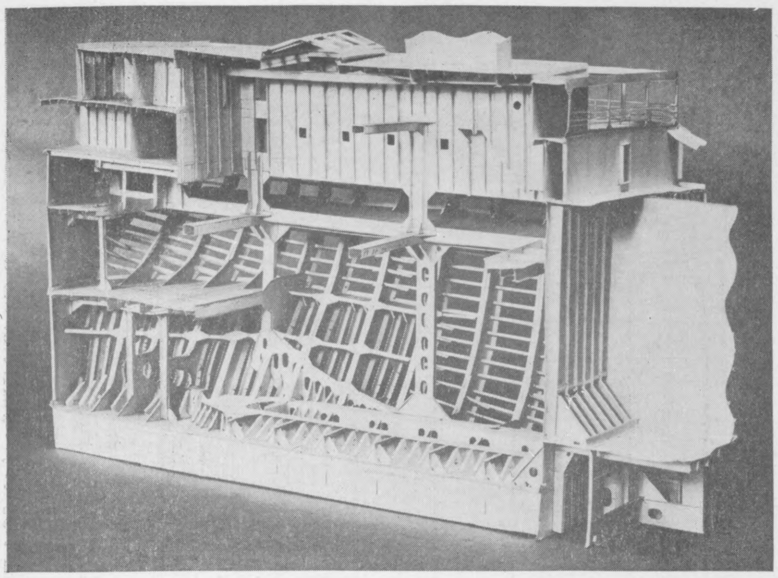
This type of ship represents the application of the Great Lakes system of ore transportation to ocean freighters. The vessels are considerably bigger than any on the Great Lakes, where the size is of course limited by the Soo locks, and they are of



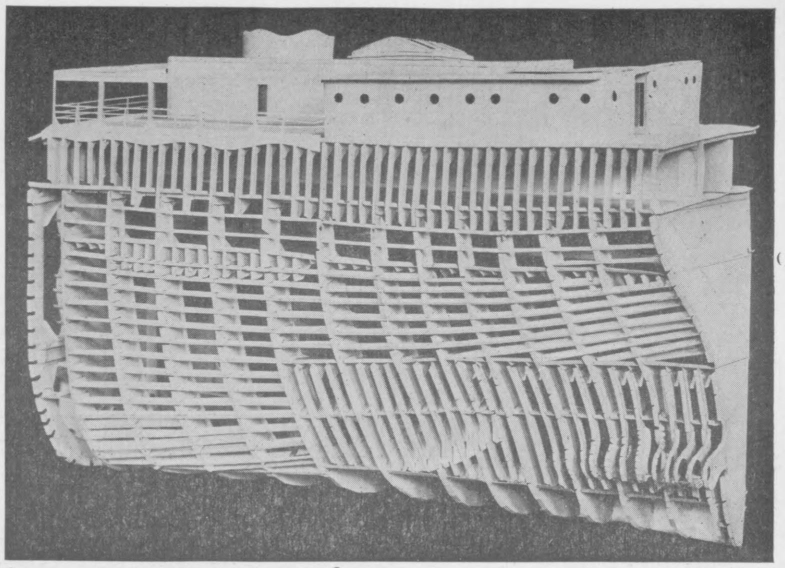
One of the main 8-cylinder 2400 s.hp. engines of the Svealand and Amerikaland



1. *View of deep double-bottom over which ore-hold is built.* 2. *Forward end of ore-hold and bow framing.* 3. *After sections of the double-bottom.* 4. *Construction of the side ballast tanks.* 5. *Decks under construction.* 6. *Perspective of ore-holds and tank spaces prior to completion of forward sections*



Model of framework in the engine room



Longitudinal and transverse framing in after peak

course seaworthy to the highest degree, which is a consideration that does not have to be taken into account on the Great Lakes. A design somewhat similar to that of the vessels in the Chile ore trade was developed for the Grängesberg-Oxelösund firm nearly 20 years ago, but it lacked the Great Lakes features incorporated in the Bethlehem design of ocean-going ore carriers and in the later SVEALAND and AMERIKALAND.

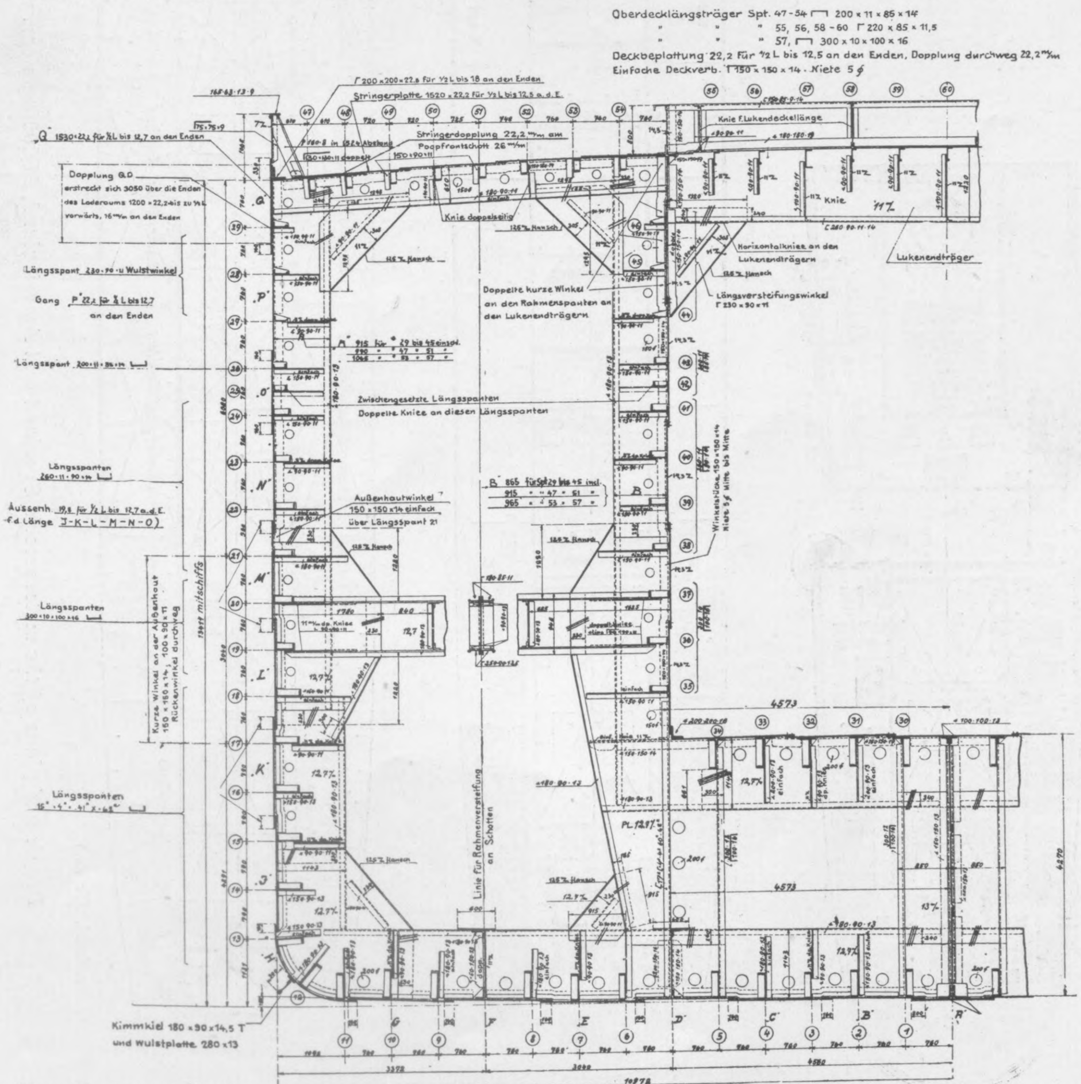
In the Great Lakes ore carriers the prime consideration has been quick loading and quick discharge. At the head of the Great Lakes the biggest vessels carrying about 13,000 or 14,000 tons of ore can be loaded and the ore-boat dispatched in about 50 minutes from the time that she moves up alongside. These same vessels can be discharged in between 3½ hrs. and 4 hrs. Rapid handling of the cargo in this way means a quick turn-round, which is an advantage to the ships and enables also the cost of handling to be reduced to an exceptionally low figure. Dispatch and discharge in such short space of time is due to the combination of big hatches in the ship with big unloading machinery ashore.

For ocean vessels the Great Lakes methods have to be somewhat modified. Whereas a Great Lakes ore carrier has one long ore hold into which the ore is dumped in piles, the ocean-going ore carrier must have its cargo firmly stowed without any possibility of shifting. In the vessels that carry ore from Chile for the Bethlehem Steel Co. the capacity of the ore holds is calculated to contain exactly a full cargo, and the holds are built into the ships like gigantic boxes in such a position that vessels can carry this heavy compact mass of cargo without being too stiff and loggy. Full width hatches extend over these ore holds for practically the full length.

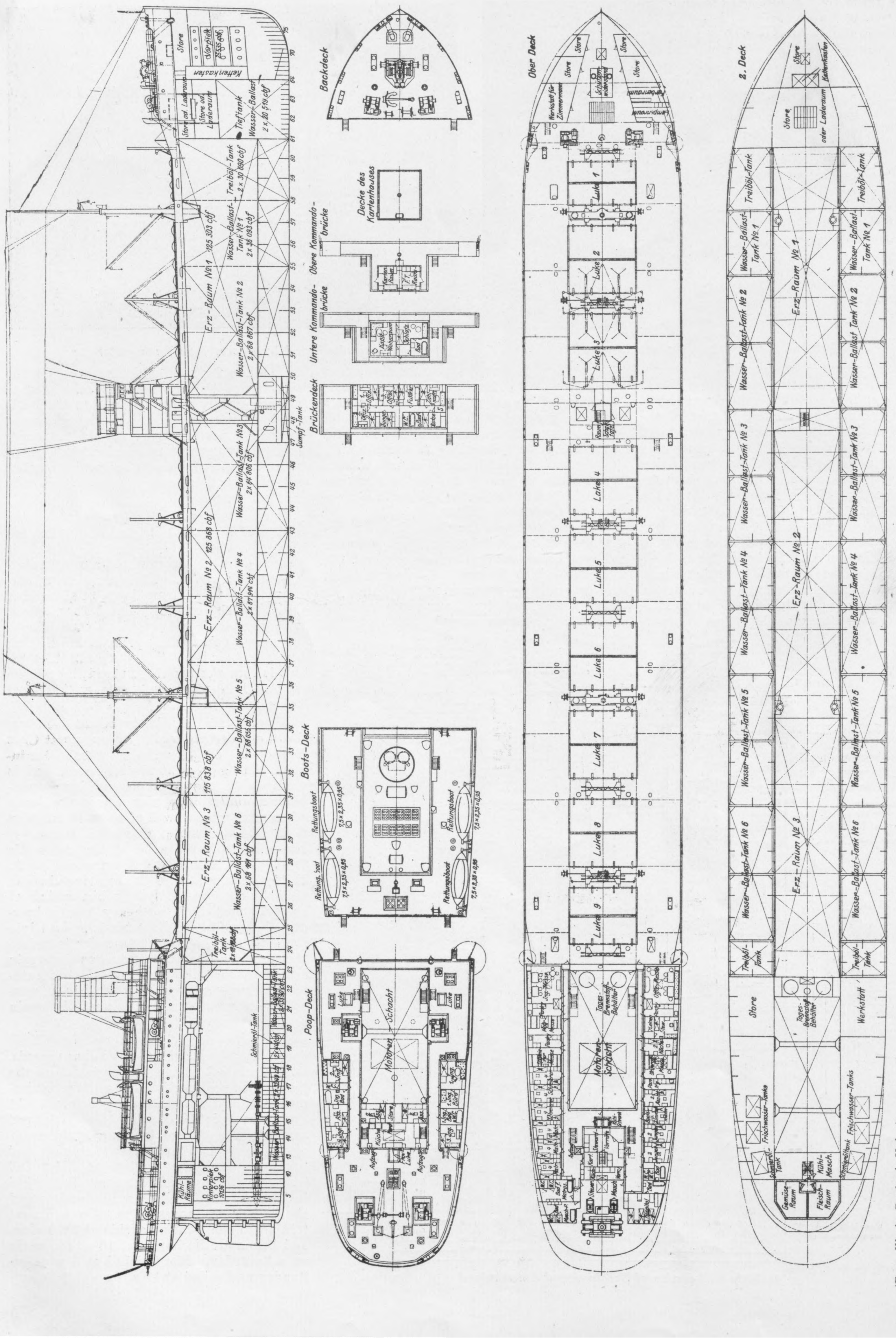
One of the greatest achievements in the design of these vessels was the combination of great strength in the deck (which is about ⅞ in. plating) with staunch, watertight hatch covers. The covers are of corrugated type and clamp down on big rubber gaskets. The covers weigh 8 tons apiece, and are raised or lowered on their hinges by power, special winches, posts and rigging being installed for this purpose. The result is that the vessels can load their cargo at practically the same rate as in Great Lakes practice.

When these big boats of 21,000 tons d.w.c. are berthed under the spouts at Cruz Grande the ore is loaded in about 90 minutes. This is a wonderful speed, and is particularly noteworthy because it is accomplished alongside a pier where the boats often pitch with a swell of about as much as 5 ft. On account of this swell the mooring winches have to be of the automatic type. The vessels load down about 32 ft. in 1½ hrs., requiring the mooring lines to be paid out constantly. A big spring of specially manufactured heavy Manila hawser of about 14 in. diameter is spliced into the steel mooring cables in order to soften the snatch on the lines, and when the strain exceeds 7 tons the mooring winches automatically pay out and they wind in as soon as the strain is relieved, until it again reaches 7 tons.

Discharge at Baltimore or New York does not approach the Great Lakes records, but this is solely due to the fact that the traffic does not justify such a heavy investment in the shore handling equipment. At Sparrows Point, Md., when the SVEALAND arrived on a typical trip north she went alongside at 12:30 p. m. on Saturday and began to discharge at 1 p. m. Her full cargo exceeding 20,000 tons was discharged in 20 hours, that is by 9 a. m. Sunday, and she sailed two hours later. On her next trip north she went alongside at 9:30 a. m. and began to discharge at 10 a. m., again on a Saturday. She finished at 5 a. m. on Sunday and sailed at 11 a. m.



(From *Werft. Reederei, Hafen*) *Midship half-section of Svealand and Amerikaland*



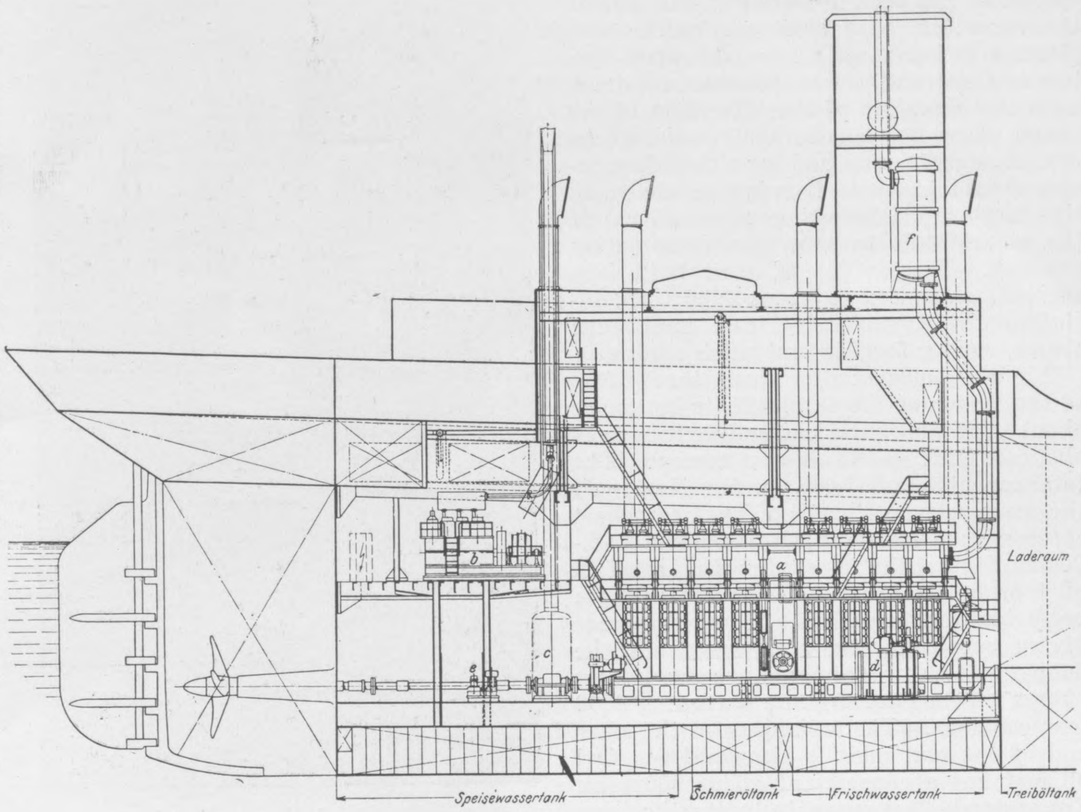
Profile and plans of the motorvessels Svealand and Amerikaland, 21,000 tons carrying capacity, 4800 shaft horsepower, equal to the biggest freighters in the world

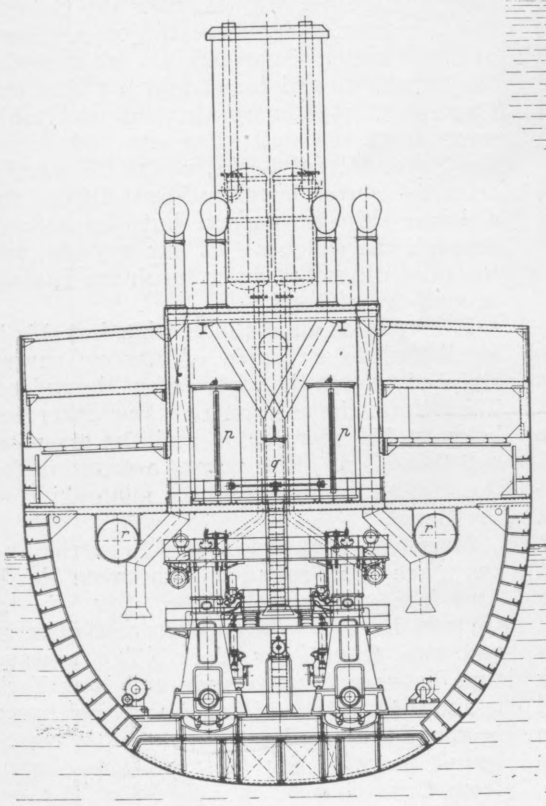
(From *Werft. Reederei, Hafen.*)

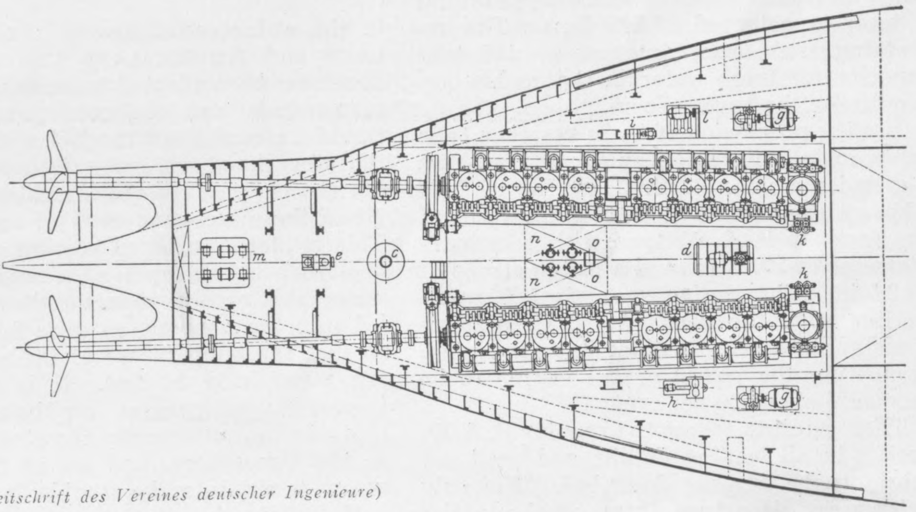
This trade indeed subjects the ship and machinery to a very severe ordeal. No time is really available for inspection of the engines in port.

In the original plans for transportation of Chile ore to this country it was contemplated that a return cargo could be obtained from the oil companies. The vessels were therefore built with oil tanks surrounding the ore holds, and one of the boats, namely the G. HARRISON SMITH now being converted to motor power is engaged today entirely in the transportation of oil. The design, however, was not so good for oil transportation as for ore transportation, and these ore carriers have had their oil tank spaces converted into water ballast space in order to obtain big tonnage deductions.

The weekly Panama Canal Record shows how effective this change has been in reduc-







— (From *Zeitschrift des Vereines deutscher Ingenieure*)

Section, elevation and plan of the engine-room of Svealand and Amerikaland

ing the tolls payable by the vessels. In the case of the SVEALAND for instance, transiting the Canal on her maiden voyage with a cargo of 20,000 tons of ore for discharge at Sparrows Point, Baltimore, there was paid on her laden transit only $4,591.20, which was equivalent to 22.956 cents in tolls per cargo ton.

This is the lowest toll per cargo ton paid by any commercial vessel that has used the Canal, although different vessels of the Ore S. S. Corp. have paid as low as 25.782 cents per cargo ton. This is due to the fact that their net tonnage measurement under the Panama Canal rules is exceedingly small when compared with their cargo capacity.

These boats would not have such an advantage if they were carrying any other class of freight. They could carry only about 8500 tons of general cargo. When the ore steamers in this trade had their side tanks arranged for the transportation of oil, their Panama Canal net tonnage was considerably higher and they consequently paid more in tolls. The Panama Canal charges are $1.20 per net vessel ton for laden vessels and $0.72 per net ton for vessels in ballast. The saving in tolls effected by the reduction in tonnage measurement due to the conversion of the oil tank spaces into water ballast space probably exceeds the profit that would have been made by the transportation of full cargoes of oil. None of the vessels in the Chile trade today carry oil, but, as has been mentioned the G. HARRISON SMITH, which is a vessel of this type owned by the International Petroleum Corp and operated in a different trade, carries nothing but oil.

These considerations had an influence on the construction of the SVEALAND and AMERIKALAND, the building of both of which was cheapened by the fact that no double riveting was required in the ballast

Engine Room Installation

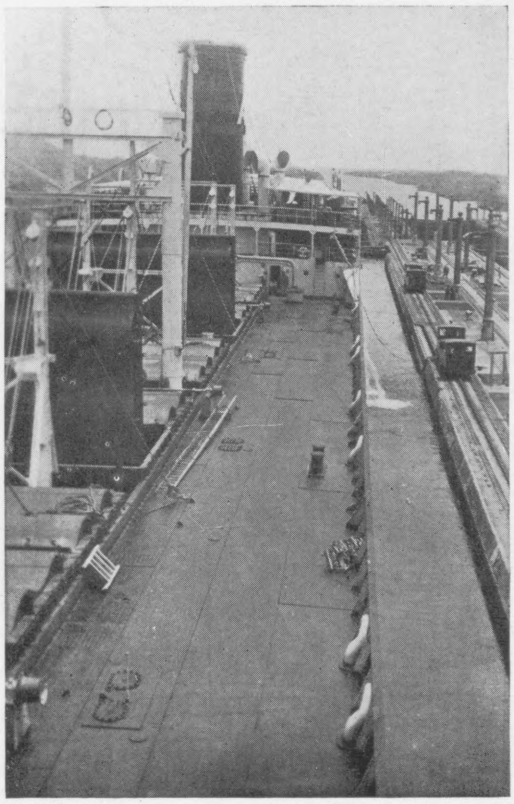
Reference	Description	No. of Units	Capacity
a	Main engines	2	2400 s.hp.
	direct connected bilge pump	1	30 tons per hr.
	direct connected sanitary pump	1	30 tons per hr.
	direct connected air compressor	1	670 cu. ft. per min.
b	Diesel dynamos, direct connected	3	100 kw.
c	Donkey boiler for heating	1	163 sq. ft., 100 lb. per sq. in.
d	Fresh water cooler and circulating pump	1	150 tons per hr.
e	Steam emergency compressor	1	200 cu. ft. per min.
f	Mufflers	2	
g	Cooling water pump	2	250 tons per hr.
h	Fuel oil trim. pump	1	100 tons per hr.
i	Fuel oil pump for daily service	1	25 tons per hr.
k & l	Bilge and sanitary pump	1	30 tons per hr.
m	Lubricating oil pumps	2	40 tons per hr.
n	Airbottles for main engines	2	9 cu. ft.
o	Spare bottles for injection air	2	18 cu. ft.
p	Settling tank for fuel oil	2	50 cu. ft.
q	Tank for solar oil	1	
t	Tanks for starting air	2	890 cu. ft.
—	Fresh water pump (not shown)	1	5 tons per hr.
—	Distilling apparatus (not shown)	1	5 tons per day
—	Rotary-transformer for electric light	1	20 kw.
—	Switchboard	1	
—	Refrigerating plant for 2500 cu. ft. cold storage	1	
—	Turning engines	2	
—	Thrust blocks	2	

spaces, as had been necessary in the case of the American built steamers which were planned to carry oil. The Isherwood system of construction was, however, retained, with the exception of the after part of the vessel where the longitudinals could not be drawn in sufficiently and were therefore replaced by transverse framing, as shown in the two very interesting illustrations of the model built to show the combination framing. The weight of steel built into the hull amounted to about 6950 tons, including the stem forging, cast steel stern frame, rudder forging and hatch covers.

The ore holds occupy about three-fifths of the length of the ship. Their bottom is about 14 ft. above the keel and their casings about 21 ft. from the sides of the hull. The total capacity is 367,000 cu. ft. Nearly all the space surrounding them is water ballast space with a total capacity of about 615,000 cu. ft., but the forward portion is a fuel oil tank of 1600 tons capacity and between ore holds Nos. 1 and 2 is a transverse space giving access to the pump room in the double bottom, where there are five ballast pumps with a total capacity exceeding 2000 tons per hour. The tank space at the after end of the ore holds is also used for fuel oil and has a capacity of about 950 tons. The water ballast space, including the forepeak, deep tank forward and afterpeak, has a total capacity of 23,500 tons. The remaining tank space amounts to 410 tons capacity for fresh water in the double bottom under the engine room.

For the propulsion of these vessels a twin screw installation has been adopted, with an 8-cylinder 4-cycle A. E. G. engine of 2400 s.hp. on each shaft. These engines are of the familiar B. & W. type with a cylinder diameter of 29.13 in. and a piston stroke of 47.24 in. The engine speed is 110 r.p.m. In line with the usual A. E. G. practice these engines are fitted for supercharging, permitting the power of the engine to be increased about 20 per cent.

These engines follow the regular B. & W. design in all leading details and need not therefore be further described. The only noteworthy departure from usual practice occurs in the engine cooling system and deserves attention. For the main cylinder jackets, cylinder heads and exhaust manifolds, fresh water cooling has been adopted. This, of course, is in order to reduce deposit



Deck of Svealand, showing hatch covers

in the water cooled spaces. In the SVEALAND and AMERIKALAND this system no doubt has been adopted because of the quick turn-rounds and the consequent lack of time for cleaning out the jacket spaces. On the other hand sea water is used for cooling the pistons, crosshead guides and jackets of the direct-connected air compressors. This is stated to be on account of the difficulty of making up for the evaporation of water that occurs when the discharges are led into the hoppers, as done in this case, so that the temperature of the piston cooling water may be felt. It is practically impossible to prevent air being carried along in the water when the circuit is open to the atmosphere, and air in the cooling water is apt to make trouble.

There are three auxiliary generator sets of 100 kw. each installed on the flat in the engine room. The rest of the installation can be followed from the legend accompanying the illustrations of the engine room.

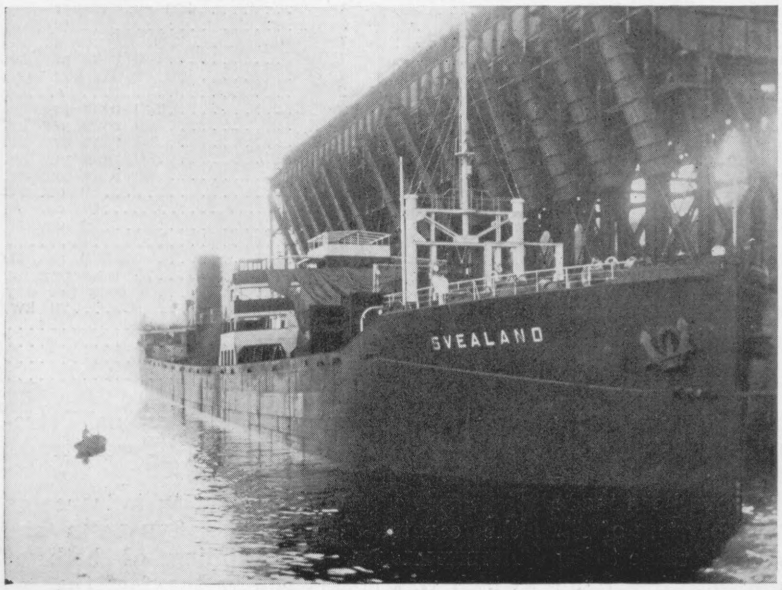
These ships have no cargo handling appliances of their own, the cargo always being loaded from spouts ashore and discharged by equipment on the piers. Beyond the hatch handling winches and automatic mooring winches already mentioned there is no other deck machinery than the anchor windlass and the steering engine, both of which are electrically operated.

SVEALAND was commissioned last May and AMERIKALAND completed her trials in July. Both vessels have a speed of about 11¼ knots at sea. Reports from the SVEALAND prove that the motorships are not only more economical than the steamers in the same trade, but that they handle better, although it is true that this superiority comes partly from the benefit of experience gained by the steamers.

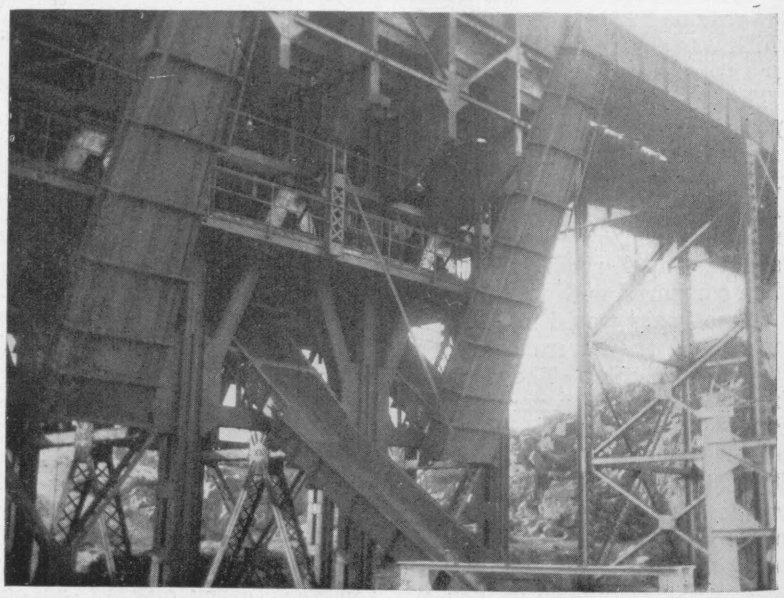
On her first transit through the Panama Canal SVEALAND was watched pretty carefully and on her way through the Culebra Cut a tug stood by for emergency. Some of the Panama Canal officials accompanied the SVEALAND and found that not only was there no trouble in working this motorship away from the wall after she had passed the Pedro Miguel Locks, where the suction from the approach wall makes it difficult for a large ship to depart, but she steered through the Culebra Cut like a yacht, and the pilot did not have to touch the engines in making the bend.

It may seem almost incredible, but this is attributed to a couple of narrow curved strips riveted to the after end of the rudder. These form the only under water difference between the motorships and the steamers and there is no other way of accounting for the greater handiness of the motorships on the Canal.

Since SVEALAND was placed in service she has made eight round trips between North American ports and Chile. Six of the voyages have been signed from Baltimore and two from New York. The average time between Cruz Grande and New York is about 19 days, and the time to Baltimore is about 7 hours less. On her last northbound passage, for instance, she cleared from Cruz Grande on Feb. 19 at 9 a. m. and dropped anchor off Quarantine, New York on March 9 at 9 a. m. Not until 5 p. m., however, was she tied up alongside the Bethlehem Steel Co.'s pier at Claremont, N. J., and meeting with further delays—understood to be late arrival of railroad



Svealand under the spouts at Cruz Grande



Close-up of the ore loading spouts

cars—she could not get away before the morning of March 11.

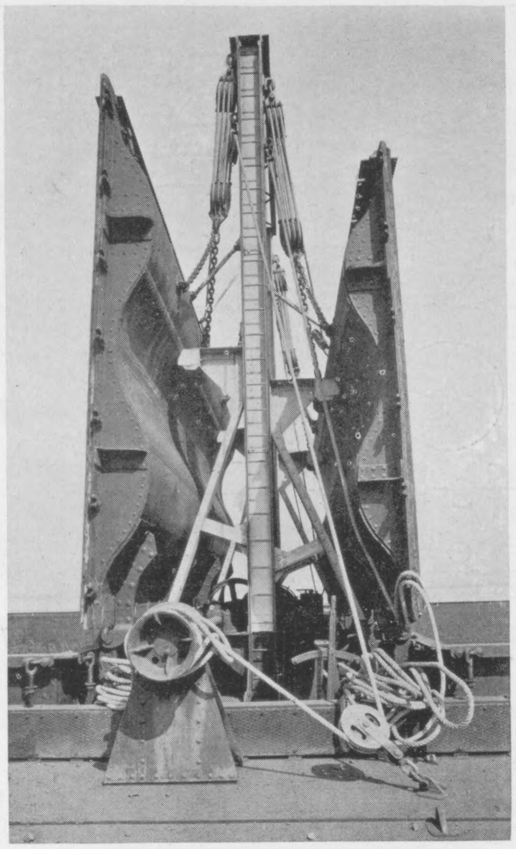
Discharge at the Claremont pier in New York harbor cannot be as fast as at Baltimore, because the Claremont pier has only two unloaders, whereas at Baltimore there are four. At Baltimore discharge can be effected in 16 to 18 hours; at New York the time is about 22 to 24 hours.

With such quick discharge and far more rapid loading, the absence of lay-overs at either terminal introduces a special problem of engine inspection and valve replacement. Only serious damage to a ship would be allowed to interefere with the fast schedule. Provision therefore has to be made for overhauling at sea.

When the chief engineer is ready to open out of the engines for inspection or for the routine change of valves he shuts down an engine for eight hours. The ship proceeds on a single screw with the loss of about 25 miles in the eight hours. The other engine is overhauled on the next trip. Thus does an unusual procedure meet the dictates of necessity.

Both these vessels make an average speed of about 11¼ knots loaded and about 12 knots in ballast. Their daily fuel consumption is about 19½ tons of 29 deg. Beaumé oil, with an average of 6400 i.hp. at 115 r.p.m. Of the daily total about 18.7 tons are consumed by the main engines and about 0.8 tons by the auxiliary machinery.

Illustrations of the unloading rig taken during discharge operations at the Claremont terminal show how exceedingly well the experience of the Great Lakes ore-carrying business have been adapted to the ocean transportation of ore. One of the big Wellman-Seaver-Morgan ore unloaders aided by a rig of older type discharges the ore as fast as the railroad cars can be brought up and moved away. The big hatch covers open up and provide the facility for rapid operation of the grab buckets. Yet these hatches are the staunchest and tightest ever sent to sea, stronger by far than any ordinary freighter has ever been equipped with. In fact, they make the old fashioned wood and tarpaulin hatch coverings still sanctioned by even the most progressive freightship owners look like the relics of a Columbus caravel.

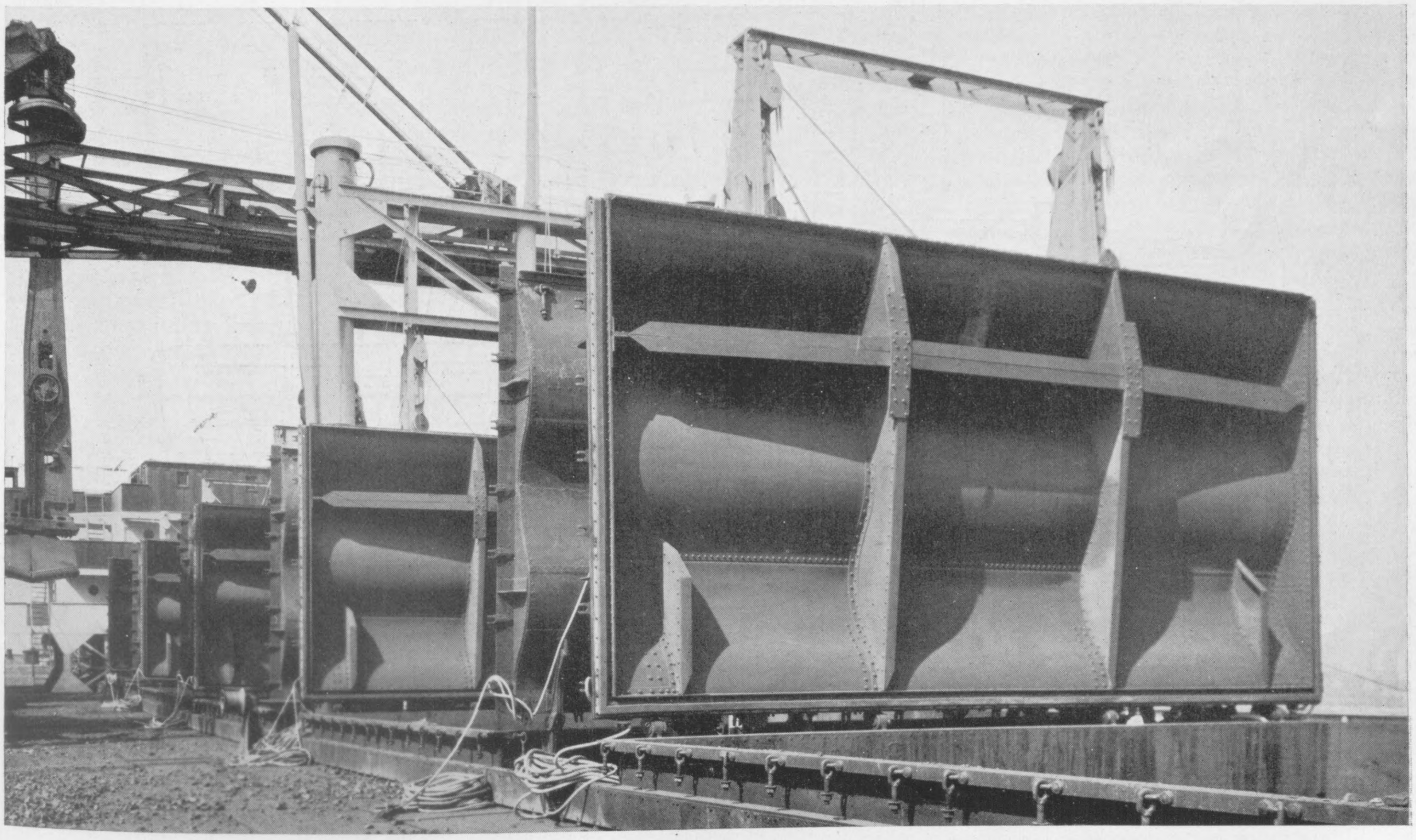


Lifting rig for hatch covers

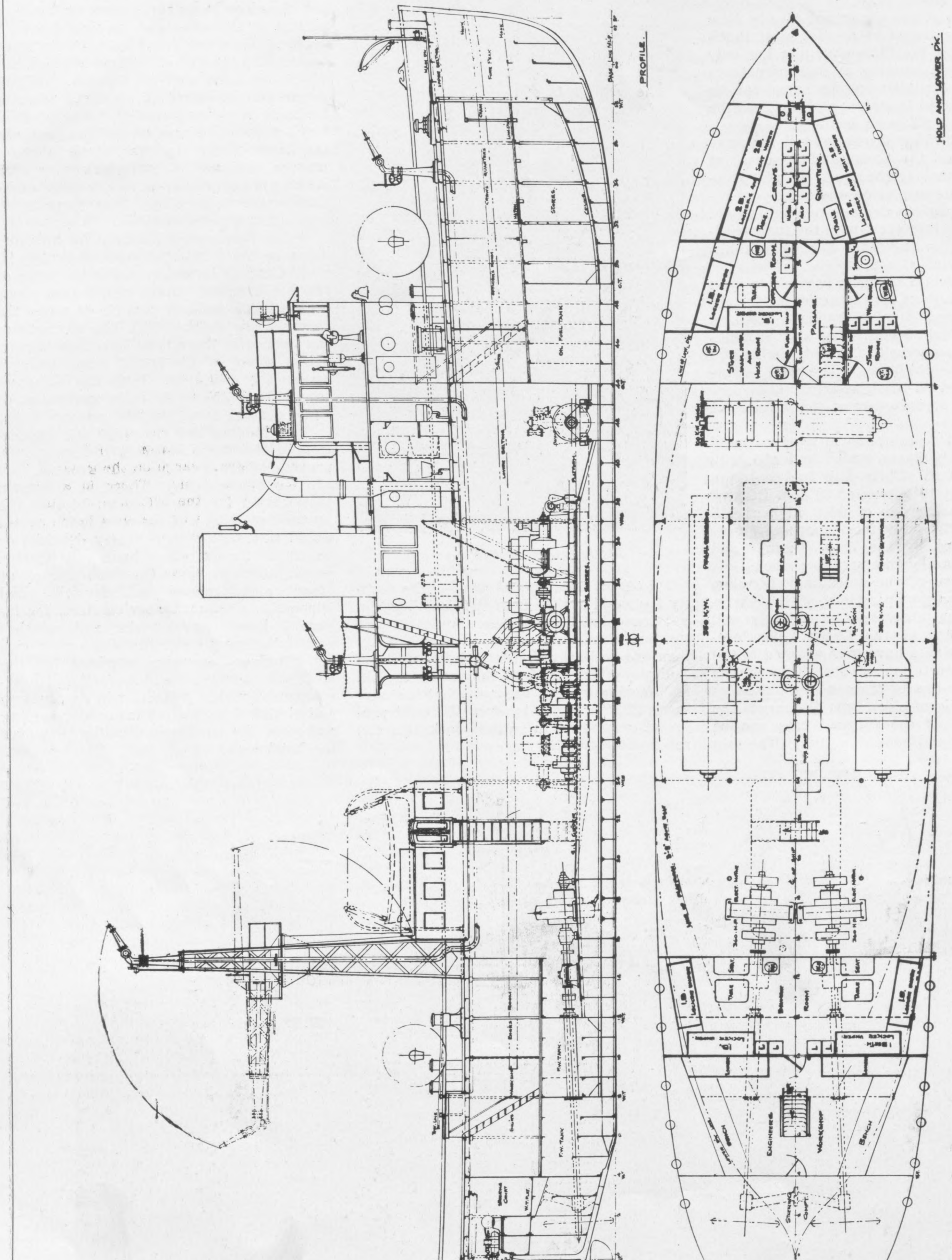
The hatches of the Ore S. S. Co.'s vessels are closed by large corrugated steel covers, hinged at the sides and opening from the center. They are lifted by ordinary hemp tackle with the aid of electric winches, one winch for each pair of hatches. All the covers can be raised in about 10 minutes. There is no encumbrance of strong backs or hatch beams or covers around the deck, and the hatch covers are clear of the hold entrances and out of everyone's way until loading is completed. A few minutes and a little careful operation allow them to be lowered into position again.

Water tightness is obtained by clamping down on thick rubber pads, about 4½ in. wide and 1½ in. thick, round the edge of the hatch cover. Bolts fitting into cleats on the hatch make it possible to screw the covers down tight. With the strength of the steel cover there is no danger of hatches being stove in by heavy seas, however severe the pounding. With the tightness round the edges there is no possibility of cargo damage from external sources.

Such hatches are criticized for application to freighters carrying ordinary case, package or grain cargo on the grounds they are too cumbersome. There is a certain justification for the criticism, because the upstanding height of the open hatch covers would interfere with ordinary derricks or winches. On the other hand only marine conservatism prevents the consideration of other arrangements of derricks and winches. Certain lumber carriers, for instance, have the derricks and winches raised, the space under the winch platforms being encased to form housings for the electrical resistances and controllers, an arrangement which protects winches, motors and electrical gear from many a swamping and gives the winchmen a better view over the hatchways.



Hatch covers of m.s. Svealand opened, giving clear access to the holds for the grab-bucket seen closed on the left



Profile and hold plan of Diesel-electric fireboat Port Houston now completing for the Port Commission of Houston, Texas

Diesel-Electric Fireboat for Houston

Powerful Fire-Fighting Vessel for Texas Port Is an Advance on All Steam and Gasoline Types

THERE is now nearing completion at the Harlan plant of the Bethlehem Shipbuilding Corporation at Wilmington, Del., a fireboat for the Port Commission of Houston, Tex., notable as the first important vessel of this class to have Diesel engines instead of steam or gasoline.

Realizing the importance to the community of having a fireboat in the Port of Houston as a result of the largely increasing shipping activities in that port, the authorities in charge made a most exhaustive study of the situation.

The naval architects, after receiving the commission to undertake the work, made a further extensive investigation of the whole subject and took into consideration what had been accomplished in the various ports of the United States in fireboat design.

As a result of the investigations, it was decided that the Diesel electric system would be incorporated into the design, this type of machinery having many decided advantages for a fireboat.

Not only is the fuel economy of the Diesel engine taken advantage of, but as the fireboat is really a power house moved from point to point by its own machinery and carrying a very powerful pumping system with it, the Diesel electric combination forms an ideal installation.

Electricity is produced by three large Diesel electric generators situated amidships, of sufficient capacity to supply all the fire-fighting appliances on the boat and at the same time to propel the boat itself. The advantages of the flexibility of this system are obvious, as at any time the current can be directed where most needed, either to accelerate the sped of the boat through the water or to produce maximum discharge of water for fire-fighting purposes, as may be desired.

Another great advantage of this design is the reduction in standby losses as compared with a steam fireboat. With the Diesel equipment, no fuel need be consumed whatever when the vessel is lying idle, except what is necessary for lighting and for sanitary purposes. Immediately upon an alarm being received, the full power of the vessel can be produced at once and she can leave for her destination ready to meet any emergency as promptly as the crew can go to their stations. On a steam fireboat unless a full head of steam is kept constantly, which is most wasteful on fuel, a very considerable time must elapse between receiving an alarm and starting to the scene of the fire. So far as gasoline is concerned, though the capital cost of the machinery is less, there are very few people nowadays who would consider the saving worth the extra hazard.

Dimensions of Fireboat

Length overall	125 ft. 10 in.
Length on water line	117 ft. 6 in.
Beam	27 ft. 0 in.
Draft	8 ft. 6 in.

The vessel is built of steel throughout, in excess of the requirements of the classification societies, which produces a rugged hull that can be worked alongside of docks or against other vessels without damage.

The hull is flush decked with moderate freeboard and rather straight sheer, and has a high bulwark extending fully around the boat from stem to stern.

The machinery is placed practically amidships, the officers and crew's quarters being on the berth deck forward, the berth deck aft being assigned to an engineer's workshop, with quarters for the engine room department. The galley and mess room are in the deckhouse forward of the stack, and above are located the pilot house and the usual navigating instruments.

At about one-third the vessel's length from the stern there is located a fire tower with a large nozzle, controlled from a platform part way up the tower, the upper portion of which, as well as the stack, is so located that the fireboat can pass under bridges of a certain specified height. In addition to this after water tower, two other water towers or turrets are placed practically amidships, one on each side, aft of the stack, these towers having upon them large and powerful nozzles and at their bases being fitted with hose connection so that streams of water may be carried at will in whatever direction may be desired.

On top of the pilot house another large nozzle or turret is mounted, and there is still another turret nozzle located on the forecastle. As an addition to these five large turret nozzles, the boat is equipped with numerous nozzles that may be shifted to any place along the rail at either side. It will be seen that fire streams can be directed at will in any direction and at various heights, so as to take account of the conditions existing in any particular fire that may require attention.

The successful determination of the machinery installation for this boat with particular reference to securing the maximum power and economy for fire-fighting purposes as well as for propulsion and operation of the boat, was very carefully studied and competently handled by Cox & Stevens, the naval architects who designed the boat and have supervised construction. After considering the various makes of apparatus in the market, it was finally determined that a combination of the Diesel engines manufactured by the Winton Engine Works and the electric equipment furnished by the Westinghouse Company offered the best solution of the problem. Accordingly, there will be installed in the engine room of this fireboat two 350 kw. Diesel electric units, and one 100 kw. Diesel electric units.

The electric power will be generated by two 500 hp. Winton 4-cycle engines, each driving at 420 r.p.m. a 350 kw. 500 volt Westinghouse generator and 25 kw. 125 volt exciter. Current can instantly be switched by the Ward-Leonard system of pilot house and engine room control from the two 360 hp. 500 volt single-armature propulsion motors to the two 410 hp. 500 volt fire pump motors, or the power can be divided between the propulsion and pump motors, or anyone or all of the motors can be disconnected from the power circuit. The revolutions of the propulsion motors can be varied and held in any speed, driving the propellers from 0 to 265 r.p.m. in either direction.

When all the power of the main generating units will be required for the fire pump motors, then an auxiliary engine driving a 100 kw. 270 volt double-armature Westinghouse generator will supply power for propulsion and maneuvering. This flexibility of power manipulation, which is not possible with any other form of drive, will enable the fire chief or pilot to maneuver the fireboat to a most advantageous position and at the same time control the flow of water up to 7000 gal. per min. at 150 lb. pressure for fighting fire.

The propelling motors are designed to give additional power so that the fireboat can be used for emergency towing purposes, thus making it possible to tow a burning ship away from adjacent traffic, piers and docks.

As a result, the authorities of the Port of Houston may be fairly well satisfied that the boat which is being built for them is in advance of all fireboats yet constructed and will be a model for the design of future fireboats.

The size and capacity of the pumps are designed to give the necessary volume to handle the following number of outlets of the sizes enumerated below:

Capacity of Nozzles

After tower nozzle, 2½ in. outlet	3100 g.p.m.
Turret nozzles (two), 1¾ in. outlet	3000 g.p.m.
On top of pilot house, 1½ in. outlet	1100 g.p.m.
On forward deck, 1½ in. outlet	1110 g.p.m.
Total number 5	8300 g.p.m.

To supply these streams the fire pumps are so arranged that they will deliver 7000 gallons of water a minute against a head of 150 lb. per sq. in. or 3500 gallons of water against a head of 300 lb. per sq. in.

The construction of this fireboat is well advanced and she will immediately upon completion proceed to Houston, Tex., where she will go on the station and will be heartily welcomed not only by the underwriters but by the shipowners and operators, as well as those having property on the water front in that port.

Cyrus H. K. Curtis' LYNDONIA, just converted from steam power to Diesel power at the Cramp yard at Philadelphia, left for the South last month without preliminary engine trials. Her new machinery consists of two 700 hp. Burmeister & Wain trunk piston engines of the same type as the single engine installed in the HUSSAR.

Lage Irmaos, of Rio de Janeiro, have ordered three twin screw passenger and cargo motorships of 5000 gross tons and of 15 knots speed from Wm. Beardmore & Co., Glasgow. The machinery will be of the Beardmore-Tosi single-acting type.

In the new tug for the Diamond P. Transportation Co., the auxiliary set will consist of a 2-cylinder 12 hp. Hill engine with an 8 kw. generator and a 2-stage Ingersoll compressor on the same base.

Metering Fuel Oil Aboard Motorships

Supplanting the Unreliable Soundings and Gage Readings by Positive Measurement Through a Meter

IN view of the almost universal acceptance of all accessories which have been tried and proven worthy of service aboard motorships it seems strange that the oil meter has not come into more general use.

Perhaps this dilatory action is the result of a lack of understanding of the merits of the oil meter. Or it may be governed to a certain extent by a feeling that to equip a ship with meters involves an expense not justified by the more accurate knowledge of the amount of fuel consumed for various purposes throughout the ship.

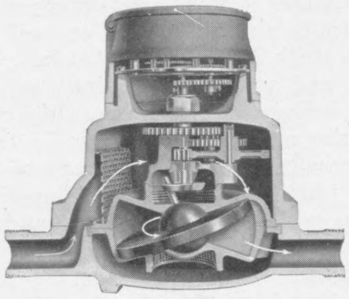
Disregarding the matter of expense involved in installing a metering system, which is not great, one should turn for a moment to the question of the accuracy of existing methods of fuel measurements. It seems incredible that the shipowner of today actually believes the figures that come into his office on log sheets at the end of the voyage. It can be more readily believed that he casts up a general average for a number of voyages and determines his rates of consumption in that way, which is at any rate quite common procedure aboard ship and not surprising when one thinks of the antiquated methods employed in many instances.

As an example of the inaccuracy of the float and gage board calibrated according to the capacity of the fuel tank, only one instance need be mentioned. On one of the Australian Commonwealth Government Line motorships the guarantee engineer, in arguing the point with the chief engineer, declared that the rate of fuel consumption was not exactly the same watch after watch as indicated in the log book, and that the watch engineers were faking figures. To prove his point he entered the engine room and switched the engines from the daily supply tank to the solar oil supply and after an hour switched back. The report coming from the engine room for that watch showed no change in the rate of fuel consumption, and the watch engineer when questioned whether any fuel had been used other than that in the daily supply tank declared most emphatically that none had.

The Diesel motor owes its existence to a demand for efficient motive power at low cost. Accuracy of fuel measurement is the only means of knowing what this cost is. The only means of being just to the engine builder, to the operating engineer, to the ship and to all things involved is to have exact figures. About the only practical method of obtaining exact figures so far as fuel consumption is concerned is to have the fuel metered.

It is impossible to obtain accurate figures on the cost of transporting cargo by sea if, for instance, the errors arising from soundings taken while the vessel is rolling are to be covered up by deductions from, or additions to, the port consumption figures, a not uncommon practice. One may blame the chief engineer for such conditions, but why blame him for making his figures balance when he must turn in to the bridge a daily report of fuel consumed and enter all such reports in ink in the log book in order to avoid embarrassing questions in the event of an accident, when his books would be "fine-combed" for any omission on his part that could be twisted against him. Surely he can not neglect his bookkeeping, and just as surely he does not know how much fuel he is consuming. His best guide is what one may call a sixth sense, a feel of the job, an indescribable means of guessing what his power plant is doing. Many a chief engineer will trust to his judgment and make entries in the log acordingly rather than attempt to obtain accurate figures from the soundings or gage board readings while the ship is rolling.

So much for the common inaccuracies of existing methods of fuel measurement. Turn to the other method and see what may be expected. Meters are precision instruments used to measure the flow of any liquid through a pipe. As a means of simple accurate volumetric measurement they have never been equaled in their field. From the domestic water meter, of which millions are in use, to the special purpose meters for measuring alcohol, hot tar, grease, chemicals, molasses—and at one time beer—their range is almost unlimited.



Section through disc type oil meter

The most common type of meter and the one which seems best suited for use in connection with the flow of fuel oil is the disc type, which has become the standard instrument in most forms of service.

Water meters have been made on this principle for about 50 years. About 25 years ago the disc meter was redesigned and adapted for use with oil, since which time its use has been constantly increasing. Oil meters really came into prominence, however, about 10 years ago with the tremendous development of oil burning and consuming apparatus of all kinds. In this development oil consuming ships have played an important part, particularly naval vessels in which the greatest degree of accuracy is demanded.

A brief description of the principle upon which this type of meter works will be sufficient to eliminate any question as to its accuracy. Reference to the accompanying illustration of a Niagara oil meter shows a measuring disc made of aluminum and carefully machined to fit the chamber in which it is placed. The nutation, or rocking around on the center ball, of this disc allows a fixed quantity of oil to pass through the chamber with each nutation. The driver pin in the top of the disc ball turns the recording mechanism.

Since only a relatively small amount of oil passes through the chamber at each nutation of the disc, gearing is required to reduce the number of revolutions to a point where it will register nearly in gallons. Intermediate gears, which run in the oil being measured, accomplish this. To prevent grit and sediment in the oil from being measured and from causing excessive wear on these bearings, they are provided, in the best meters, with dirt proof jewel bearings automatically lubricated as the result of running in oil.

To insure accurate registration one more set of gears is placed in each meter between the intermediate gears and the registerer, called change gears. These have a variable number of teeth, and by using different combinations the registration of a meter can be speeded up or slowed down by very small steps. Before leaving the factory each meter is tested by running oil through it, and the proper set of change gears is installed to make the meter registerer accurate.

There may be a prevailing opinion that the mechanism is so intricate that in the event of the failure of the meter to register accurately nothing can be done about it unless it be returned to the factory. Such is not the case, however, for if at any time while in use the meter should register inaccurately, as for example the viscosity of the liquid to be metered is changed, it merely indicates that a different set of change gears is needed. By sending to the manufacturer a report of the nature and extent of the trouble, together with the serial number of the meter a new set can be calculated and sent to the user, in most instances without cost.

Lest the extent of such inaccuracies be misinterpreted it is best to explain that they are seldom encountered in the ordinary range of viscosity of fuel oils used aboard motorships. That is, they will keep well within 1 per cent of accurate if the range in fuel viscosity is not greater than from 20 deg. Bé. to 35 deg. Bé. and the temperature range does not fall below average sea temperatures nor rise above 200 deg. F. In most instances no such great ranges exist, and accurate results may therefore be expected. However, in the event of a fixed change a constant may be calculated which, when employed in connection with the meter reading, will bring the figure to the correct amount.

It is readily apparent that no such constant can be calculated to take into account the vagaries of fuel tank soundings taken while the ship is rolling or gage readings taken under similar circumstances.

Despite the advantages thus far enumerated the meter has certain inherent fallacies which should also be taken in to account when one is contemplating their employment. One, and perhaps the most discouraging one at that, is the fact that

(Continued on page 286)

Repairs from the Chief Engineer's Viewpoint

A Broad Review of the Shore Influence and of Port Operation as They Appear to the Ship's Engineers

By A. B. Newell

NO piece of machinery will profitably suffer abuse, and motors are no exception. For this reason I am making no special point of motor repair. Motorships have invaded the steamship's realm, are being operated by the same firms and are being repaired at the same yards as the older type of vessel. They are suffering or profiting, as the case may be, by the result of years of experience of ship operation.

Unfortunately there exists a difference between the items commonly included in repairs to a motorship's machinery and that of a steamship's.

We commonly refer to annual surveys as repairs when as a matter of fact much of the work of conducting surveys on motorships has to be done when repairs are not necessary and merely to comply with rules of the various super-conservative classification societies. It is gratifying to note that less and less of this extra work is being demanded each year.

At any rate, whether a piece of machinery is taken apart for survey or repair the work must be done with care, and the following paragraphs refer to work on all types of vessel with equal force.

Marine operating engineers do not have the same point of view with regard to this all important matter that executives of steamship lines do. We commonly hear from the latter in terms of dollars and cents which constitute the business man's everyday trend of thought, but seldom hear from the engineer whose single code of morals is of a mechanical nature, impelling him to treat his machinery well and let the cost take care of itself.

The engineer may talk freely to his peers, but not to his superiors. Yet it might be better for all concerned if he had a greater power of self advertisement. I propose to express his viewpoint, being influenced greatly by everyday conversations with the general run of engineers and not forgetting my own experiences.

Too often, right at the eleventh hour of a repair period we find the engine room filled with shore mechanics, looking for something which cannot be found, screwing a nut on a bolt which needs no nut, sliding around greasy floor plates, getting in one another's way, dozing on a pile of blocking or swearing because the long hours of overtime have just about knocked them out. We have all been through this and know it to be uneconomical.

When we think of the way it started it is in a sense amusing. It would be funny if it were not so discouraging to think that the work is not being done right. It is not a lack of good mechanics, but altogether too many of them: so many in fact that they cannot work profitably.

The ship came into port running fairly well, but the time for a general overhaul or survey was due, and the chief engineer had allowed a number of items of work to accumulate to be done at this time. With his papers in nice order his repair list topped the pile.

The port engineer came aboard and pounced upon this list, scanned it with an eagle eye and made some quick and sensible decisions *sur le champ*, as the French say, for being a practical man he dislikes the mad scramble at the last hour just as much as does the chief engineer. He is human of course, and has weaknesses which may cause him to blue-pencil 50 lb. of rags or the repair of the steward's coffee percolator, but in a general way he is very capable and often feels that the whole weight of the steamship division of the company rests upon his shoulders and depends upon him for success. We will not antagonize him, for we may want him to give us a job some day.

This is the first day in port. The port engineer will pass judgment upon the matters of lesser importance, but must consult the marine superintendent about other ones which may involve an expenditure of a few extra dollars.

He in turn may have to consult the manager, and if there is a really big item some of the directors must get their fingers in the pie, or so it seems to the man on the ship who has waited two days to get a start made on the items which he feels to be of most importance and which he knows would require less than half an hour of typing and telephoning to get in hand.

In the meantime estimates of the cost of the jobs will be given by local repair yards, along with promises of absolute satisfaction in quality of workmanship and time required to finish the job, most of them bidding low on the contract items to get a chance at the extras.

Once the work is started there seems to be no great rush, for every one, with the exception of the chief and port engineers, seems to feel that since the ship is in for repair she will be tied up indefinitely and every one will have plenty of time to finish everything.

Not many men came aboard the first and second days, but perhaps not many had been hanging around outside the gate of the repair yard lately. More will be picked up in another day or two. Good mechanics too—regulars at the gate.

By the eighth day we are nervous about finishing on time, for only ten days have been allowed in which to complete the work, and overtime is taboo unless we get stuck.

The purchasing agent has been warned time after time to get the stores aboard before the last day, so he puts forth a special effort and orders them delivered the afternoon of the next to the last day. The shipchandler fell down on the job, and stores came the morning of the last day, for shipchandlers seem to have a means of knowing when ships will sail in spite of anything the owners may think.

Overtime has been sanctioned and a gang is on all night. Fuel comes on a barge at two in the morning.

The fight is on against time of sailing and finishing the work in hand. Engineers get tired, some of them quit. The steward kicks up, and will not set out a night lunch for the men working overtime, and everyone is ready to fight.

It appears as if we will never finish in time, but we do, even though a hot crank develops at the last minute and a crosshead pin, neither round, square nor oval, is pounding as the tugs help into the stream.

Every one on the dock draws a long sigh of relief as the ship rounds the point and passes out of sight and out of mind. We hear them remark: "Well, she is off and won't bother us again for a month or two."

The port engineer glances over the assembly of port captain, port steward, purchasing agent, stevedore boss and repair shop foreman and does not voice an opinion. He strolls into his office shaking his head and thinking of the times he has sailed just that way. He knows the ship will go and come perhaps safely enough, but he does not like the way things have gone. He may be heard to mumble: "I'll be damned if I know what to do about it."

We do not blame poor mechanics, incapable engineers, indifferent port engineers nor executives of the line for this condition. The repair yards are not making fortunes at this sort of thing, and all in all when we talk the problem over with men most interested it is agreed that just such a condition as this exists in all save a few ship operating lines.

We know the machinery suffers, and motors perhaps suffer more than steam machinery, for the average mechanic does not understand them as well as he does a triple-expansion engine.

A remedy is needed and we would like to know the secret of the company which can get a ship off in first class condition without the mad scramble at the last hour.

There are a few who can. We have our own opinion as to how they do it, but will not express that just now.

A Diesel electric towboat for the Long Island R. R. Co. is to be built by the Staten Island S. B. Co. and will have two 400 hp. Ingersoll-Rand engines with Westinghouse electrical equipment.

A 2700 s.hp. McIntosh & Seymour single-acting engine for the Shipping Board was scheduled to complete its 30-days non-stop full power trial in the last days of March.

As a factor influencing cost of production a number of cylinders for one size of big engine which Sulzer's have in hand in their Winterthur shops at the present time is interesting. With the recent addition to their order list of the engines for the new Grace motorliners Sulzer's now have on hand 52 cylinders of 26.77 in. diameter. This is equivalent to over 30,000 i.hp. in one cylinder size.

COMMODORE Arthur Curtis James' barque ALOHA, after conversion by the Staten Island S. B. Co. from steam auxiliary power to Diesel electric auxiliary power, was rushed to Florida, her passage South constituting the machinery trial. She has Winton Diesel engines and Westinghouse electrical equipment.

Carl Fisher, the real estate developer, plans to build a big ferryboat for operation from the eastern end of Long Island, where he is developing a property of 10,000 acres. This boat will be propelled by the two 500 hp. Winton engines from his yacht SHADOW K.

A fleet of Diesel electric ferryboats for automobile and truck traffic across the Hudson River in New York City is being organized.

Drawing attention in the British House of Lords to the government loan made to the Silver Line under the British Trade Facilities Act, Lord Parmoor complained that this was diverting money to promote competition with British traders, for though the ships were to be built in Great Britain they were to be used in connection with what is really an American line in competition with British shipowners and for carrying American products to the Far East.

An increase of 14 vessels of 51,503 net tons last year brought the total of Swedish motorships up to 228 with 205,255 net tonnage. The steamer tonnage decreased to 700,005 net tons.

Three Army Sternwheelers

GILLETTE, the first of three Diesel-electric stern wheel towboats for use by the Army Engineers will be put in service this month. She and the BURNETT are 70 ft. long and will be assigned to the Cincinnati District; the third, the GOUVERNEUR, is 100 ft. lond and will be utilized in the St. Louis district.

The GILLETTE and the BURNETT are being built by the Nashville Bridge Co. at Nashville, Tenn., with General Electric propulsion equipment consisting of an 85 kw. generator set supplying power to a 100 hp. motor which drives the stern paddle wheel through chains. The exciter is direct connected to the main engine, and in addition to supplying excitation, provides current for the operation of electric capstans, pumps, refrigeration, steering gears, etc. Both boats will be used principally for dredging and lock service in the Cincinnati district, covering the Ohio River and some of its tributaries.

The GOUVERNEUR, the 100 ft. boat, is being built at the Howard Ship Yard, Jeffersonville, Ind., and will be put in service this Spring. The propulsion equipment for this boat also was built by the General Electric Co. Two engine-driven generator sets are provided, each having an output of 90 kw. and supplying power to a 225 hp. motor which, in this case, drives the stern paddle wheel through gears and pitmans. Direct-connected exciters are provided for both excitation and the operation of auxiliaries. This boat will be used for dredging and lock service in the St. Louis district on the Mississippi River.

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Readers are invited by the Editor to submit articles, photographs or drawings relating to motorships, marine oil-engines or auxiliaries. Contributions used in the magazine are paid for on the 15th of the title month of issue, and other contributions are returned as promptly as possible.

Three Diesel yachts were floated out of the dry basin and a Diesel dredge was launched down the ways at the Newport News S. B. & D. D. Co.'s yard at Newport News, Va., on Mar. 20 last. The yachts were Richard M. Cadwalader, Jr.'s SAVARONA, Edmund S. Burke, Jr.'s JOSEPHINE and Hugh J. Chisholm's ARAS. The dredge was the RAYMOND for use of the Engineer Corps on the Delaware River. Leading characteristics of all these boats have been detailed in previous issues of MOTORSHIP.

Two 800 hp. Bessemer engines are to be installed in Carl Fisher's fast yacht SHADOW K., in place of her present two 500 hp. motors.

An order for 62 Diesel engines of 150 hp. each has been placed by the U. S. Coast Guard with the Winton Engine Co. for installation in 31 patrol boats of about 125 ft. length.

Four single-screw motorships of about 6000 tons d.w.c. each have been ordered by Elder Dempster & Co. of Liverpool from Harland & Wolff. These vessels are for the West African trade.

Capacities of pumps and other engine room auxiliaries are often rated at speeds considerably inferior to the rated speeds of electric motors, and in such cases the motors driving them have to operate below rated power. For instance, a pump needing 5 hp. to drive it may have to be direct connected to an electric motor of 15 hp. or 20 hp. rating, because the pump has been designed to operate at a much lower speed than the motor. In the case of the EAST INDIAN'S pumps listed on page 202 of the March MOTORSHIP for example, the horsepower quoted was the actual power required from the individual electric motor and was not the maximum rating of the motor. These figures were furnished by the Sun S. B. & D. D. Co., and, as explained in the article, are not the rated powers of the Westinghouse motors driving the various pumps. A shipyard has to select a motor that will give the required power at the required pump speed irrespective of the normal rating at best motor speed. The Westinghouse Elec. & Mfg. Co. has drawn our attention to the possibility of confusion arising in the minds of readers due to the differences between pump ratings and motor ratings, and we are glad to explain the matter here in more detail than was done last month, in case the explanation previously given was not clear enough for all.

Metering Fuel Oil Aboard Motorships

(*Continued from page* 284)

it is limited in its capacity to measure a flow of oil. The flow must be rapid enough to cause the desired fall in pressure as the oil passes through the meter, and which is required in order to bring about the regular nutation of the disc.

The minimum rate of flow that can be measured accurately with this type of meter is about 6 gal. per hour, and there are a number of ships equipped with auxiliary engines requiring less fuel than that. Therefore it would be impractical to attempt to meter the flow of oil to an engine so small.

Having this in mind, it is readily apparent that the one mistake which could be made without much trouble would be to judge the size of the meter by the size of the pipe connections leading to and from it, in place of taking into account the capacity of the instrument itself and thus install a meter which would not work purely because it had a capacity too great for its service.

On the other hand there is practically no limit to the high rate of flow, and varying pressures (not differences between inlet and outlet) have no appreciable effect upon the functioning of the mechanism, provided not more than 300 lb. per sq. in. is employed (and with specially constructed cases as much as 500 lb. per sq. in. can be applied without causing damage). Such pressures are of course far beyond the capacity of any ordinary fuel transfer pump or filling line used for fueling a ship.

A very simple installation, which would cost less than $50 would be to install a meter between the tanks and the fuel transfer pump and another between the daily supply tank and the main engine or engines, thus accurately measuring the fuel placed in the tank and that going to the propelling power, leaving a difference which could be considered to be an accurate measurement of the fuel consumed for auxiliary purposes.

The method of application, however, should need no explanation. What seems more to be needed is a thorough understanding of just what the meter is and such advantages as may be expected in the way of accuracy of data obtainable.

Operating Experiences on 2-cycle Engines

How Slight Changes in the Mechanical Arrangements Gradually Helped to Overcome All Minor Engine Troubles

By Dan C. McKay*

IN an article in the MOTORSHIP of July, 1925, dealing with Krupp 2-cycle single- and double-acting engines the author thereof expressed a fear that the heat shields are liable to corrosion on the upper side when a fuel oil of sulphur content is used. I can state from 2½ years' experience on the m.s. LOKI, which is equipped with 2-cycle Krupp engines of an earlier design than those of the W. A. RIEDEMANN (ex-ZOPPOT) that not one of the 12 heat shields replaced was defective in the location mentioned. A year and a half after going into service the W. A. RIEDEMANN had not experienced any operating difficulties from these shields, the original ones still being in use, but I have no information about her later operation.

The m.s. LOKI is one of the oldest motorships afloat, the hull and engines having been built by the Germania Werft at Kiel, Germany, in 1912. She made several transatlantic voyages prior to the war, at the outbreak of which she was seized and interned in England, where she remained for the duration of the war. After the armistice the English government allocated her to the Reparations Committee, which turned her over to the Standard Oil Company of New Jersey to operate, along with the m.s. WOTAN and three steamers, all of which, except the WOTAN, are now being operated under the Interallied and American flags by the above named company.

LOKI was towed from England to the Kiel yard owned by Krupp's, where her main engines were thoroughly overhauled and practically an entire new set of auxiliary machinery installed.

An uncommon feature of this installation is that there are no attached compressors on the main 2-cycle propelling units which are rated at 1250 i.hp. each. Injection air and starting air are supplied by auxiliary compressor sets, of which there are three direct connected with Krupp 4-cycle units developing 210 i.hp. each, to each of which is also attached a 75 kw. compound wound generator for lighting and auxiliary machinery duty.

Two auxiliary sets are kept in continuous operation while the ship is under way, and as one generator is sufficient for ordinary conditions the compressor loads are balanced to equalize the total loads on each engine. Fifteen days constitute a normal run of one auxiliary engine, and 10 days are therefore allowed for the overhauling of the standby set.

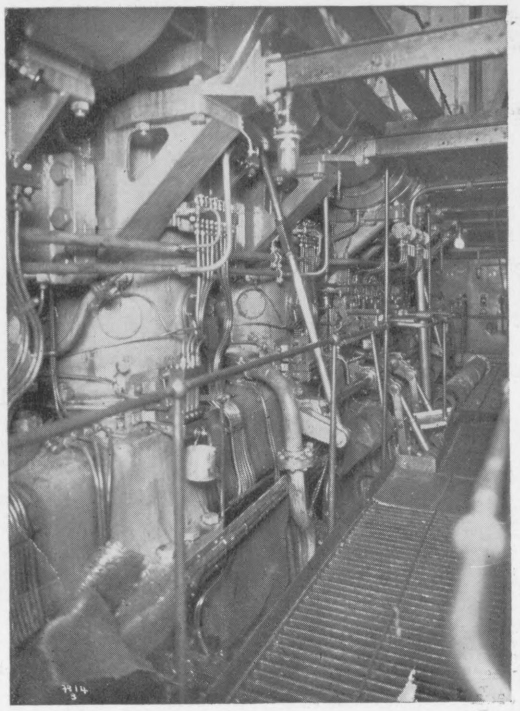
The advantages of such an arrangement on a tanker requiring quick turn-rounds in port is readily apparent. All the compressor overhauling is done at sea, the entire crew being available in port for main engine repairs.

The l.p. and i.p. compressor suctions are controlled by piston valves mounted on a common stem, eccentrically driven from the crankshaft. These valves require no attention for long periods of time. All the other compressor valves are of conventional design.

Each compressor is rated at 353 cu. ft. of free air per min., but the actual capabilities are slightly less. All three compressors are in use during maneuvering periods, the bypass and bleeder valves being located at the maneuvering station, which makes a very handy arrangement.

Scavenging air for the main engines is supplied by two double-acting pumps, beam driven from the crossheads of Nos. 3 and 4 cylinders of each engine. The pumps are located above the floor plates on the outboard side. To the under side of the beams are attached the main engine circulating (two for each engine), bilge and sanitary pumps. The circulating pumps supply water for the cylinder jackets, heat shields and, as later arranged, also for the pistons, delivering at about 30 lb. per sq. in. to the chambers and thence by-passed at about 12 lb. per sq. in. to the water jackets. The surplus is used for the pressure system of piston cooling at about 10 lb. per sq. in.

Scavenging air is delivered to the header at between 2.4 and 2.8 lb. per sq. in. and admitted



Fuel pumps on lower grating—m.s. Loki

to the cylinders by two scavenging valves located in the head and actuated by the main camshaft through suitable rocker arms. The scavenging pumps are slightly small, however, and 5 lb. per sq. in. would be a more desirable pressure for those engines. Back pressure in the exhaust pipes runs from 1.2 to 1.8 lb. per sq. in., so the necessity of more scavenging air is apparent—the exhaust is always colored at full power.

Starting is carried out by groups of three cylinders run on air till the other three of the same set pick up the load, when all cylinders are put on fuel. This change is carried out by two hand levers, controlling air rams connected to the rocker arm fulcrum shaft. The rocker arms are mounted eccentrically, and when the levers are put in the starting position the starting valves only are in operation, the rollers of the fuel valves and of the scavenging valves being then entirely clear of the cams. In the stop position the camshaft is put either ahead or astern by two turns of a handwheel. A reducing valve on the main starting line admits air at about 350 lb. per sq. in. When the engine starts rolling, three cylinders are put on fuel and immediately followed by the other three, whereupon the automatic starting valve not only shuts off the starting air, but also drains the line between the reducing valve and the engine. As the maneuvering levers are moved from the running position to the stop position the injection air from the respective cylinders is automatically shut off and the injection line past the admission valve drained.

At Kiel in 1923 the original cylinder heads were replaced by those especially recessed for heat shields. The shields are secured by the inlet and outlet cooling water pipes, which pass through holes in the head, and are fastened by drawing the chamber firmly against the head by means of special nuts which also act as packing gland followers. On the opposite side of the shield from the pipes one anchor bolt is loosely secured to a lug on the chamber, and a copper ring around the bolt is drawn hard up against the head to make the joint for this connection, packing being dispensed with.

Originally the shield cooling pipes were packed with about 15 small piston rings, and distance pieces were furnished over which they fitted. These were a source of continual annoyance, because the pipes surging and expanding and contracting chafed the metal rings and were worn through in several instances, necessitating a complete removal of the shields. Invariably the packing pieces had to be reamed out of the head, a shield renewal taking as long as eight hours in some cases. Other cases showed that sulphurous acid had been at work, the combustion gases having free access around the packing rings and water being obtained by the inlet pipe sweating. However, the clearance space between the head and shield being accessible to a free circulation of the hottest gases and the shield not being connected with the head so that a transfer of heat could take place at this point, I believe that the temperature of the clearance space is very little, if any, below that of the cylinder space, thus eliminating the possibility of moisture in liquid form on the surface of the shield.

By dispensing with the old type of cast iron shields and substituting cast steel shields and by packing the inlet and outlet pipes with common H.P. steam rod packing, all shield troubles have been eliminated. Shields now need to be changed only due to scale stopping the flow of cooling water and causing the shield to crack. Removal takes only 2½ hours, the shield coming out very easily with soft packing.

This arrangement seems to be one solution of 2-cycle engine difficulties because a considerable number of spare shields can be carried, whereas the cost and space required for cylinder heads would be prohibitive.

A slight difference exists in the arrangement of the cooling water lines of the LOKI and W. A. RIEDEMANN. On the latter vessel the outlet lines are led down the outboard side of the columns and discharge into the bilge. On the LOKI the pipes are led up to an inspection tank located above sea level, and the water gravitates overboard, thus relieving the bilge pumps of excessive duty.

Each system has its advantages. In the LOKI system scale and dirt are much more liable to clog the outlet pipes, restricting the flow of water and accentuating the liability of overheating and cracking. A higher pressure also must be maintained to force the water up to the inspection tanks. To eliminate the obstructions to the water flow, an air line was connected up to each shield inlet pipe, and

*Chief Engineer of the m.s. LOKI for 2½ years.

upon signs of trouble the affected cylinder was cut out, the water shut off and the shield blown through with 80 lb. of air pressure.

Another departure from ordinary practice is found in the crosshead bearing lubrication system, each crosshead carrying its own pump. The pump housing is secured to the pin, while the varying angularity of the connecting rod actuates the lever connected to the plungers. For each side these pump housings are supplied by sight feeds or a gang oiler system.

The main lubricating oil is contained in two sump tanks located under the floor plates, into which the lubricating oil drains from each end of the engines. Attached gear pumps draw the oil from here for delivery to the main bearings and timing gears, those being the parts under pressure. The cranks are oiled by gravity feed into cups identical with steam practice.

Part of the oil taken from the sumps is bypassed to the settling tanks located in the muffler room. The oil enters the bottom of one tank and as it rises becomes heated from the intense muffler heat. Much of the water and accumulated foreign matter settles to the bottom of the tanks. As the oil rises to the top of the first tank it overflows into the bottom of another, when it passes through the same performance. As it overflows from this tank it is led either back into the sumps or to a No. 600 De Laval purifier and thence to the sumps. Drain lines in the bottoms of these tanks are led to the purifier also. This system not only greatly assists in extracting much of the sediment and water, but also furnishes an economical means of preheating the lubricating oil before centrifugal purification.

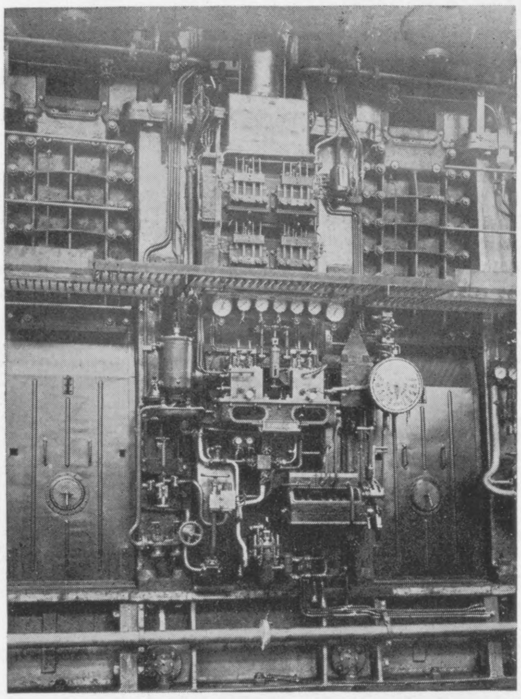
The lubricating oil consumption of any open base Diesel engine with a gravity system is necessarily high. The average daily consumption of the oil for all purposes is about 35 gal.

Each piston carries its own piston cooling pump, salt water being the cooling medium. A check valve was contained in the inlet pipe and the telescope pipe furnished the plunger. Both telescopes worked in a subdivided chamber, one side constituting the inlet and the other the outlet. After enormous difficulties had been experienced with salt water leakage past the telescope packing and into the lubricating oil systems, the chambers were removed from the inside of the column and fitted on the outside of the engine splash plates. Subsequently all the valves were removed, after trouble had been experienced with sticky and worn out checks. Water is now supplied by the main circulating pumps, and a special connection made to an auxiliary pump.

This eliminates all possibility of checks sticking and of pistons getting overheated. A more uniform temperature can be maintained at the outlets and considerable noise is eliminated. This system has given perfect satisfaction.

Timing gear trouble we experienced the first few voyages. The main transmission gear is fitted in two halves to the crankshaft coupling between Nos. 3 and 4 cylinders. This gear has 48 teeth and meshes into a gear on the vertical shaft having 32 teeth. With a ratio of 3-2 in the gears, the vertical shaft turns at 1½ times engine speed, the camshaft speed being attained by a 26-tooth gear on the top of the vertical shaft meshing with one of 39 teeth on the camshaft. Formerly the large gears were steel and the smaller ones bronze. After changing the order and reducing the friction on the bronze gears, very little trouble has been experienced.

A novel steering gear about completes the departure from conventional practice. It is a Krupp designed gear working on the oil hydraulic ram principle, two parallel rams being connected through connecting rods to the rudder stock crosshead. Air maintained at a constant pressure of 120 lb. per sq. in. in a tank also used for the maneuvering rams of the



Engine control stand on Loki

main engines, is the controlling element. This is led to a manifold, the valves of which are actuated by several cams mounted on a horizontal bar worked by a common type of telemotor. Air admitted and released by these valves is led to, and from, a piston connected by a common rod to the oil control piston valve of the steering rams. The pressure is directed on the forward side of one ram and on the after side of the other, and vice versa for the opposite working of the gear. The ends of the ram piston not under pressure are open to discharge back into the main oil supply tank.

The motive power is derived from two sources, a centrifugal pump directly connected to No. 3 auxiliary compressor engine being used when this unit is in operation and an independent motor driven pump being installed in the steering house to be used during the stand-by period of No. 3 auxiliary engine. Both sets are used as a precautionary measure while entering or leaving port.

When the telemotor centering device brings the steering gear to rest the oil is by-passed back into the supply tank. This is a simple, rugged and reliable steering device, and has proven itself in all kinds of weather.

Turning the main engines at 100 r.p.m. gives a loaded speed of about 9.5 knots and light about 10.25 knots under favorable conditions. The fuel consumption for all purposes, exclusive of donkey boiler is about 76 bbl. daily.

A 3-fire Scotch boiler is installed for cargo pumping and one small vertical boiler is used for heating purposes only. The warping winches and anchor windlass are also steam driven, as also is one general service and one fuel transfer pump. A 20 kw. steam driven generator is for lighting and pumping duty while in port and also for the motor driven 2-ton Brunswick ice machine.

Ferryboat Gains by Centrifuging

Treatment of Fuel Oil and Lubricating Oil Has Effected Big Economies

FOR nearly four years now the Diesel electric ferryboat POUGHKEEPSIE has been in operation. She has two 135 hp. Winton Diesels turning at 450 r.p.m., driving 90 kw., 230 volt Westinghouse generators with 9 kw., 125 volt exciters direct connected to the main units. The electric power drives two 100 hp. motors. W. F. Corp has been chief engineer aboard her for about three years. On a recent visit to the boat we inquired about his experiences with the fuel oil and lubricating oil separators, which were installed after difficulties had been experienced in the early operation of the boat, and he has furnished the following report.

"During the summer of 1923 we had several minor machinery troubles caused mainly by lubrication. We were using an average of 15 gal. of oil in 16 hours, which is a day's run. We changed oil and made other changes which cut it to 8 to 9 gal. and in the late summer to 7 gal. During all that time we had quite a bit of bearing trouble and also a lot of dirt in fuel oil which caused trouble.

"In September, 1923, this got so bad we were about to have to tie up, when I went down to the De Laval Separator Co., located here in Poughkeepsie, N. Y., and had a No. 200 Purifier demonstrated and ordered one at once. In two days it was in operation, and since that time we have not had a minute's trouble from dirty oil. It worked so well on fuel oil that I connected it up to purify lubricating oil, because it was in use only about three hours a day on fuel oil.

"Before that time we were pulling pistons and grinding valves every six weeks. The results were so good that in December we ordered another No. 200 Purifier and changed to D. T. E. Extra Heavy lubricating oil. Since that time have cut the oil consumption from 2½ to 3 gal. a day, grind valves about every three months and pull pistons every six months, at which time we find only one or two rings to an engine stuck. Previously we would find three or four on a piston.

"With D. T. E. oil we go over the bearings twice a year. There are some that we have not been able to take up for a year and a half, and very seldom over three 1/1000ths. out of any.

"The 100 hp. motors are connected to Falk reduction gears through Falk-Bibby couplings and drive 4 blade propellers of 5 ft. diameter at approximately 200 r.p.m. The thrust is taken care of by Kingsbury thrusts and they have only been adjusted once in two years. The air compressors, lubricated with D. T. E. X., have not been pulled for two years.

"The POUGHKEEPSIE is 140 ft. x 52 ft., of the hull-fin type with four gangways and carries 36-40 autos. She operates on a 30-minute schedule between Poughkeepsie and Highland on the Hudson River, New York. The distance is about ¾ mile. The schedule is maintained twelve months a year if the ice permits. She will go through seven inches of solid ice without stopping. In the last three years we have been off the run very little.

"The engineroom is ventilated with a 30-inch wall blower of 8200 cu. ft. per min. and discharges out. In summer when we can open the ports and run the blower at full speed the engine room is cool and free from all gases and smoke."

An interesting example of the abrasive material that can be taken out of fuel oil was reported recently from a motorship. The fuel oil centrifuge yielded a quantity of reddish material that looked like wet brick dust. If this had not been taken out of the fuel oil before use it would have passed through the engine, and no great imagination is needed to picture in the mind how material of this sort contributes to the wear in the engines.

West Coast Systems of Airless Injection

Design, Construction and Operation of Pacific Coast Types of Airless Injection Diesel Engines

CONSIDERABLE progress has been made on the Pacific Coast in the manufacture of the Diesel engine of small powers, and, although the industry is comparatively young, it has become quite an important branch of engineering. Several firms once engaged in gasoline engine construction have developed a design of this engine principally for marine purposes.

These small engines are all of the airless injection type and are rapidly supplanting the gasoline engine in boats engaged in the Pacific Coast fishing industry. Several engines have also been installed in towboats, yachts and ferries, and owners and operators alike are enthusiastic in their praises of these small power plants.

All of the engines are of the 4-cycle type, made in sizes from 60 b.hp. to 300 b.hp. averaging 300 r.p.m. and carrying a fuel consumption guarantee of 0.45 lb. per b.hp.-hr., a figure often reduced in practice. Fuel oil of 24 deg. Beaumé gravity is generally used.

Much controversy has centered round the respective merits of the air injection and airless injection engines, but for the needs of the class of boats these engines are being installed in, there is no question as to the suitability of the airless injection type.

Elimination of the air compressor and its auxiliaries is an important feature, and the reduced weight and overall length are factors not to be overlooked. The combustion is quite as good as in the air injection type, perfectly clear at full load and with just a slight color when the engine is idling. The cylinder compression required is not so high as in the air injection type, 350 lb. per sq. in. being quite sufficient for starting under cold conditions. Through the fuel valves used a high degree of atomization is obtained, and the fuel changes rapidly from a liquid to a gaseous state, promoting the necessary turbulence required to obtain complete combustion.

All engines are of the closed crankcase type, employing force feed lubrication. The lower portion of the bedplate is used as a sump or settling tank. Two lubricating oil pumps are employed, one taking the oil from the sump tank and discharging into a small gravity tank. From this gravity tank the oil flows to the suction of the force feed pump, the discharge of which is led to the main bearings, through the drilled crankshaft to crankpin bearings, then up the connecting rod, which is drilled its entire length to the wristpin and returns to the sump tank. A duplex strainer is fitted in the lubricating oil system, allowing the operator to change over and clean while running.

In the types using push rods for lifting the valves, the camshaft is inclosed in the crankcase and obtains its lubrication by splash. The engines employing overhead valve motion, transmitted through vertical shaft and spiral gears, have a camshaft of hollow steel tube construction, into the center of which is led a pipe from the force feed system. A small hole is drilled from the face of the cam in the center of the camshaft, and centrifugal action delivers through the small drilled hole to lubricate the cam faces, rocker arm rollers, pins, camshaft bearings and spiral gears, from which it is led to the governor parts and back into the bedplate. The cylinders and pistons are lubricated separately by means of a mechanical lubricator.

The lubricating system is very economical, the oil fed through the cylinder mechanical lubricator being considerably more than is used through the force feed system. In fact the cost of running a 65 hp. engine for fuel and lubricating oil is 12½ cents per hour.

The main bearings are all fitted with removable shells, babbitt lined, to facilitate a quick repair if necessary, spares being carried for this purpose.

A few of these engines are direct reversing, but the majority are fitted with clutch and reversing gears. Many of the boats are one man jobs, the controls for clutch and engine regulation being carried by extensions to the pilot house. It speaks well for the reliability of these engines that the operator, after once starting the engine, only looks in occasionally to see how it is running. The operators are mainly recruited from former marine gasoline engine operators. In the larger engine rooms former steam engineers have charge.

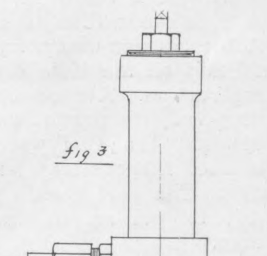
Fuel Oil System

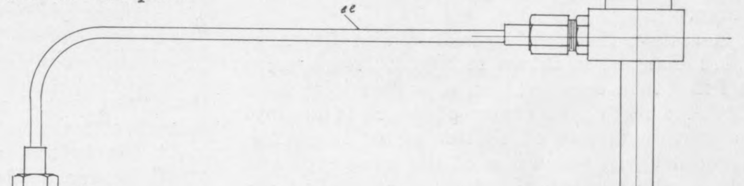
The construction of the high pressure injection system closely follows the same general principle in all Pacific Coast engines, although each has its own patented features. The particular system selected for description embodies several novel features and is under complete governor control at all speeds and powers.

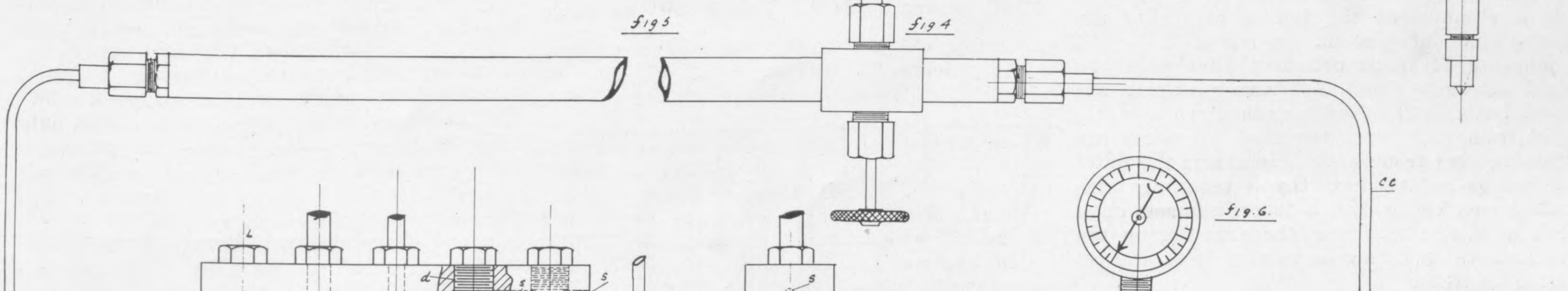
The fuel pump body is made from a steel block machined all over and drilled to accommodate the various parts assembled therein.

The plungers j and k are connected to a cross-head (not shown), to which a driving rod is fitted to actuate the suction valve tappet rod, in the same manner as on air injection engine pumps, the suction valve being held from its seat during a portion of the plunger stroke. The lower end of this tappet rod f is fitted to an eccentric, the rotation of which by the governor raises the suction valve off its seat during a greater or lesser portion of the plunger stroke.

The low pressure plunger cylinder k is fitted at t with a pipe Tee, into which are secured







Figs. 1, 2, 3, 4, 5 and 6, showing the relation between the essential details of a typical airless injection system

commercial horizontal suction and discharge valves. The suction side is piped directly to the main fuel oil tanks and the discharge side piped to a gravity tank above the engine. The overflow from the gravity tanks is returned to the main fuel tanks, the gravity tank always being maintained at one level or full. From the gravity tank the oil flows through the pipe *ff* and enters the fuel pump through the low pressure pipe fitting *x* and four drilled holes in the suction valve cage *z*. On the upstroke of the high pressure plunger *j* the oil is drawn past the suction valve *a* into the chambers *p* and *p-1*. The downstroke of the plunger ejects the oil through the discharge valve *b* into the strainer *h* by means of the drilled passage *s*.

The strainer *h* is of sufficiently fine mesh to insure that no particles of foreign matter of a size liable to stop up the minute holes in the fuel valve nozzle are allowed to pass into the system.

From the strainer compartment the oil is forced through the high pressure pipe union and pipe *g* to the accumulator, Fig. 2, the pressure contained therein being registered on the high pressure gauge, Fig. 6. From the accumulator the fuel oil is led by means of the pipe *cc* to the fuel oil manifold, Fig. 5, which is fitted with a needle valve, Fig. 4, to each engine fuel valve, Fig. 3. From the end of the fuel oil manifold another pipe *dd* is led to the under side of the fuel pump body at *n*.

This section of the pump body contains a relief valve set at about 5,000 lb. per sq. in. to insure that no excessive pressure occurs in the system. When this relief valve lifts, the escaping fuel oil delivered past the valve is returned to the main fuel tanks by means of the pipe *gg*.

The pressure maintained in the fuel oil system for full power conditions is 3,500 lb. per sq. in., and although this pressure may seem high, there is really no danger attached to it, for it is confined in small section steel tubing 0.125 in. inside diameter, and the accumulator and fuel oil manifold are also of steel tubing construction. A hand pump *o* is fitted in this pump body to prime the system for starting purposes, the fuel valve being temporarily rendered inoperative by means to be described later. The fuel system can be primed up to 3,500 lb. per sq. in. by means of the hand pump.

A section through the suction and discharge valve chamber is shown in Fig. 1. Each valve is fitted in a cage held on a gasket joint by a threaded nut *c* ,the center of which is squared to provide means of tightening or removing. The discharge valve *b* is of the wing type and has a retainer at the top to centralize the spring *bb* as has also the cap nut *d*.

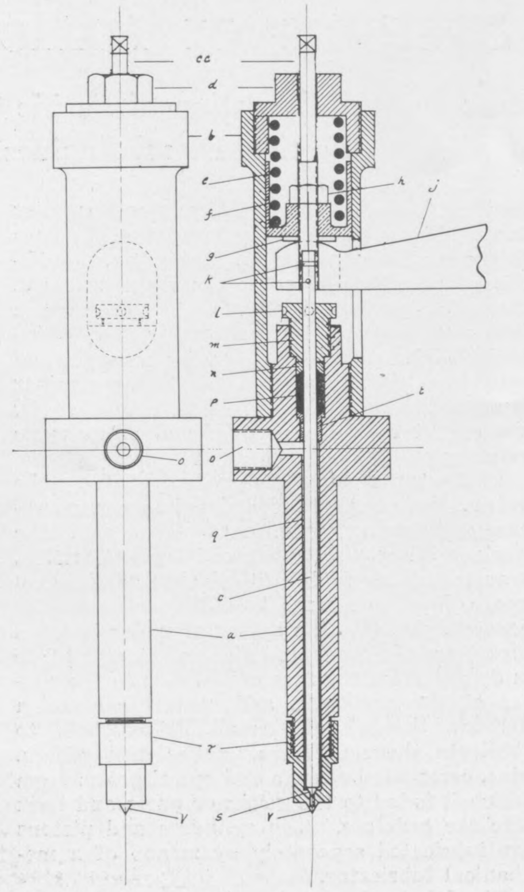
Engines of three cylinders have only one high pressure pump, but two plungers are fitted for a four cylinder engine.

In the pump here described all parts are contained in one assembly. In others the relief of fuel is accomplished by a manually controlled overload valve, a lever working on a rack quadrant releasing the pressure on the overload valve spring or raising it as the occasion requires.

Fuel Valve

In Fig. 7 is shown a sectional view of a fuel valve. It is simpler in construction than the fuel valve used in the air injection engine. The drilled oil passages in the body of the valve, the atomizing tube, plates, distance pieces and the milled cone at the end of the atomising have all been eliminated, while the flame plate at the end of the valve has been substituted by a nozzle drilled with minute holes.

The body of valve *a* is drilled its entire length, slightly larger than the needle valve diameter, forming an annular space or clearance *q* through which the oil is led from the high pressure union fitted in *o*. This annular



Figs. 7 and 8. The fuel valve

space carries the full accumulated pressure, and on the needle valve *c* being lifted from its seat by the fuel cam action through lever *j* the fuel is forced into the combustion chamber through the minute holes drilled in the angular end of the nozzle *r*, the high pressure (3,500 lb. per sq. in.) causing complete atomization and instantaneous combustion.

The needle valve *c* and *cc* is made of two sizes of steel, the lower portion *c* telescoping into the upper portion *cc*, and, secured by means of pins *k* to *cc*, is the spring guide *f* carrying the spring *e*. At the lower portion of *f* is a hardened steel washer *g* in direct contact at lifting periods with rocker arm lever *j*. The upper portion of *b* is threaded to receive the spring nut *d* and the lower enu threaded to secure to the valve body *a*.

At the bottom of the packing gland is a small bronze guide *t*. This is all the guide provided, the exceedingly small valve lift and the flexibility of the needle valve ensuring the needle valve seating concentrically. The gland nut and washer are seen at *m* and *n* respectively and a square is provided on the needle valve end to rotate the needle valve when working. The lower end of the nozzle nut *t* contains the angular valve seat *s*. Below this seat is a small drilled hole into which are drilled the minute spraying holes round the periphery of nozzle end *v*.

Cylinder Heads

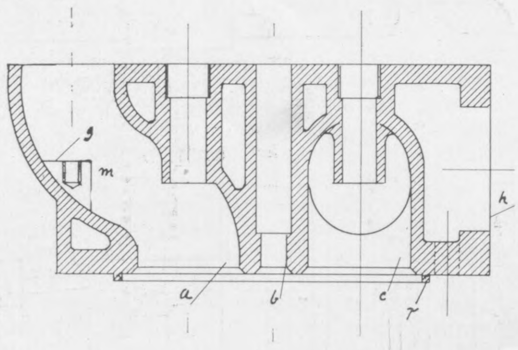
The cylinder head shown in cross section in Fig. 9 and in plan view on Fig. 10 is a type generally used by a well known manufacturing firm.

Much trouble has been experienced in the past in heavy duty gasoline marine engines by the circulating water being restricted by deposits in the cylinder heads. When engines are run in boats tied to wharves at low tides or in shallow rivers a great deal of silt is drawn into the circulating system, and again when the engine is stopped in a heated state after a long run, if it is not cooled down by an auxiliary water pump, precipitation takes place in the water jackets. In the course of time, unless they are cleaned often heads and cylinders will crack.

In the type of cylinder head shown ample means have been provided for cleaning purposes. At *h* are shown three square flanges which can be easily removed, allowing accessibility to the entire circulating water system in the head. These holes also greatly facilitate the foundry work, enabling the cores to be held firmly thus reducing the liability of their moving from place.

The air inlet valve port is seen at *a*, the air being drawn through the air suction silencer (shown in Fig. 13) into the cylinder through the cored passage *m*. The air suction silencer is secured to the boss *g* by a long stud *s* passing through its entire length and held in position by a nut on top of the silencer. At *b* is seen the fuel valve port and the exhaust is at *c*. The air starting valve is fitted at *e* and the flanged pipe for it is secured to the boss *o*. The exhaust manifold is secured to the oval flange *k*.

In Fig. 12 is shown another method of ataching an exhaust manifold. By this means any cylinder head can be easily removed without loosening all the manifold bolts.



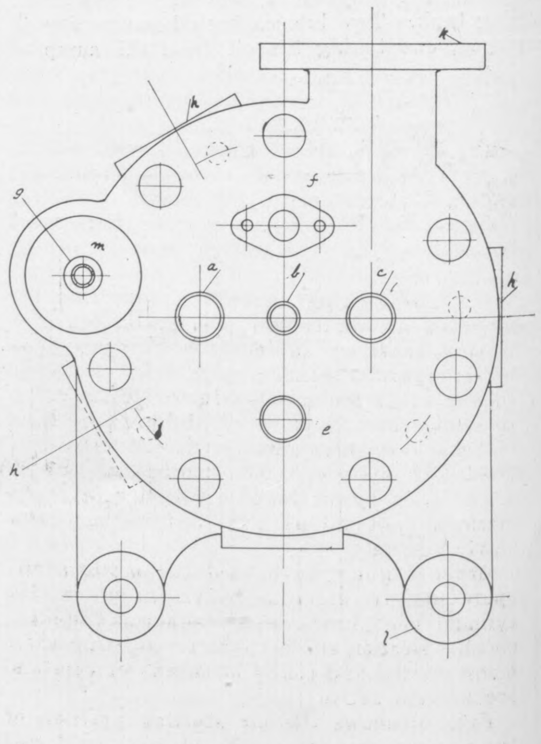
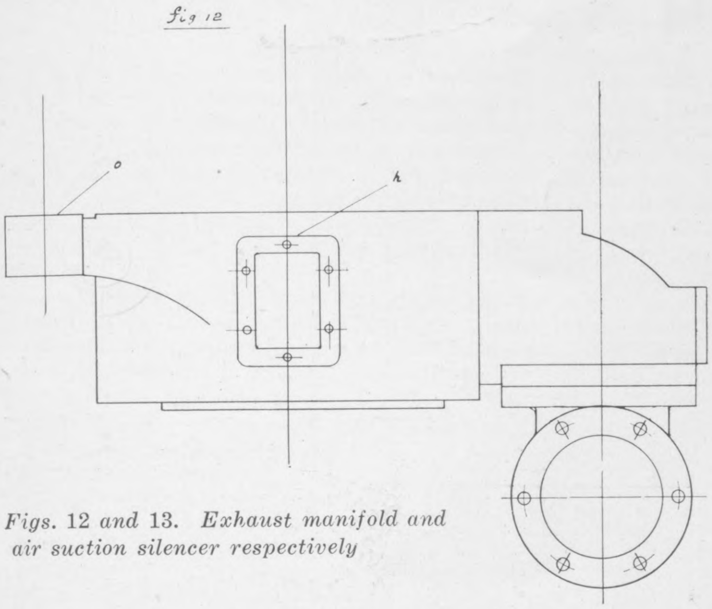


Fig. 9: section through cylinder head. Fig. 10: plan of head. Fig. 11: water connection

Fig. 11 shows the water connections, the lower end of which is threaded into the cylinder top, a clearance hole being drilled in the cylinder head and a water tight joint assured by a rubber gasket *p* pressed into the clearance between the cylinder head and cylinder top. The gasket being of soft rubber does



Figs. 12 and 13. Exhaust manifold and air suction silencer respectively

not prevent the sealing of the head and cylinder by means of the gasket joint ring *r*. The pillars for mounting the rocker arm eccentric controls are fitted on the bosses *l*. On other makes, a cast by-pass pipe fitted on the outside of the cylinder head and cylinder is used to carry circulating water from the cylinder to the head. The circulating water discharge is fitted to the oval base *f* on top of the cylinder head.

The exhaust and inlet valves are shown in Figs. 14 and 15. The steel valve stem *d* is threaded into the cast iron head *f* and riveted over. The bronze valve stem guide is seen at *e*, the threaded portion screwed into the cylinder head. The tension of spring *c* is held by lock nuts *b* and *a*, the top face of *a* being hardened for the rocker arm depressing screw or roller. It will be noticed that the exhaust valve head is fitted with a recessed apron *g* to protect the spindle from the exhaust flame when the exhaust valve is open.

Air Starting System

Starting the engine is effected by means of compressed air, a pressure of 150 to 200 lb. per sq. in. being carried in the tanks for this purpose. A small auxiliary compressor, gasoline driven, is used to charge tanks in emergency.

The engine is fitted with a small bore single-stage compressor driven from the crankshaft and sufficient to keep tanks charged to 150 lb. when the engine is running. A relief valve is fitted in the air discharge lines to prevent overcharging.

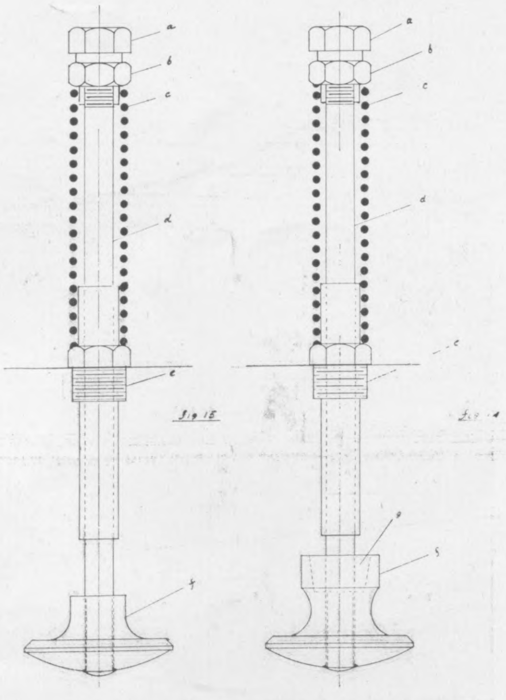
For air starting one well known firm employs an air distributor, located on top of a vertical shaft, an automatic valve being fitted in the cylinder head, a pipe being led from each port in the air distributor to its corresponding valve in the head, the pressure holding the automatic valve open and delivering air to the cylinder. The timing and cut off are effected by means of a rotary valve automatically put out of action when the engine starts on fuel.

Other engines have used the mechanically operated air starting valve fitted in the cylinder head, in the same manner as employed on air injection engines, the air starting valve being opened and closed by means of cam and rocker arm action.

Fig. 20 shows the air starting position of the exhaust cam and push rod roller on dotted line *a* and the running or fuel position of roller on line *b*. To start the engine on air it is barred around into position until any air starting valve is opened then the compressed air tank main valve is opened, admitting air to the engine.

Starting is effected by the movement of quadrant lever *c*, Fig. 16, into the boss of which is fitted the end of an eccentric shaft *d*. This eccentric shaft runs through each rocker arm and the ends are carried in the pillar *f* which acts as a bearing. It has a different degree of eccentricity in each rocker arm to accomplish the desired results of air starting and running on fuel, i.e., in the air starting position the fuel valve and air inlet valve push rod rollers are lifted clear of their respective cams and valves, and rendered inoperative, while at the same time the air starting valve cam and auxiliary exhaust cam are brought into action.

The air starting cam shown in Fig. 19 has two lifting faces, giving an air starting impulse every revolution, as has also the exhaust cam, thus allowing the spent starting air to be exhausted every revolution through the exhaust valve. This action, as will be readily seen, also acts as a compression release during the air starting period.



Figs. 14 and 15. Exhaust and inlet valve

When the quadrant lever *c* is in the position shown in Fig. 16, the exhaust cam is in the fuel or running position and the auxiliary toe of the exhaust cam running clear, but when the lever *c* is brought from *n* to *nn* the eccentric motion brings the roller *g* down towards the cam and places the auxiliary exhaust cam and air starting cam in the air starting position.

The eccentric action of moving this lever c is shown in each cam, the solid line representing the running position and the dotted line showing the air starting position. Assuming the lever *c* is at the postion *nn* on the quadrant, the fuel valve and air inlet valve are inoperative and the engine runs as a 2-cycle air engine for a brief period until sufficient momentum has been gained. When the lever c on one cylinder is moved on quadrant *h* to the position *n*, this movement by its eccentric action places the air starting valve and the auxiliary exhaust cam out of action, brings the fuel and air inlet valve into action and the cylinder immediately begins to fire on fuel. The lever *c* is rapidly moved over on the other cylinders and the main valve to the starting ranks then closed. The whole operation of starting from cold until all cylinders are firing takes about 8 seconds.

Cams

A simple design used for the fuel valve cam is shown in Fig. 17, a detachable piece of tempered tool steel *p* being fitted into a cast iron disk *l*, which in turn is secured to a boss *m* secured to the camshaft. Elongated holes are carried in the disc as shown, to allow for setting the fuel valve, and when the desired setting has been attained, a fitted bolt is placed at *o* to prevent its alteration.

The air inlet cam Fig. 18, air starting cam Fig. 19, and exhaust cam Fig. 20 are of cast iron with chilled faces.

Remarks on Engine Starting

In preparing to start the engine, it is advisable to go over each system separately and assure oneself that everything is correct on one system before going to the other.

First, place the clutch in neutral and release the compression in the cylinders. Then bar the engine over two full turns to make sure everything is free. One reason for this is that, although the engine may have been all right when last stopped, it is quite possible that water leaks into a cylinder overnight, and to attempt to start under a condition like that would probably mean a big repair job. Open up all valves to and from the engine on the circulating water system. See that all valves and strainers are open in the force feed lubrication system.

Next, place all control quadrant levers in the air starting position—this closes any fuel valve that may be held open by its cam. Open up all needle valves on the fuel manifold to the fuel valves and set the overload valve about 3,000 lb. Prime the whole fuel system to this pressure by means of the fuel hand pump. The engine is then ready to start when the air is admitted to it.

If the engine is cold the pressure in the air starting tanks should not be less than 150 lb. per sq. in. If the engine is heated from a recent run, 80 lb. should be sufficient to start. Bring the pressure in the tanks up to the required amount by the use of the auxiliary compressor if necessary. Close the compression release and open up the main starting air valve. The engine will immediately run on air.

As quickly as possible move the control quadrant lever into the fuel or running position, one cylinder at a time, until all are firing. Then shut off the starting air, and

by means of the overload valve control bring the engine to desired speed and power.

Watch carefully all pressure gauges and temperatures at reasonable intervals. Keep a pressure on the lubricating oil of between 5 and 10 lb. per sq. in. and a discharge circulating water temperature around 125 deg. F.

Failure to Start

If the engine fails to turn over quickly enough with starting air, examine and correct the air starting valves. If the air starting valves are all right and the engine refuses to start on fuel, take out the fuel valves and clean the nozzles with a small wire cleaner. Fill the nozzle with coal oil and exert a downward pressure on top of the nozzle with the ball of the thumb. This will cause the oil to be discharged through minute holes in the nozzle and readily seen.

Another cause of failure to start is loss of compression, due to piston rings leaking or exhaust or inlet valve leaking. This should be corrected at once, for if the compression is too low the temperature is not sufficient to ignite the fuel. Sometimes an exhaust valve spindle is found to be tight, not allowing the valve to work freely. In such a case pour coal oil on the spindle and work the rocker arm until it is free.

Loss of Power While Running

If the engine shows signs of falling power while running with a smoky exhaust, probably one or more holes are stopped up in the nozzle. Find out the weak cylinder and change the nozzle as quickly as possible.

To Prime Fuel Pump

When the feed pipes to the fuel pump have been disturbed or the suction and discharge valves removed for any reason, it will be necessary to prime the pump to discharge all air from the system.

First, see that all air is removed from the feed pipe leading from the fuel gravity tank, by allowing a full column of oil to flow through it. Couple up to the fuel pump and assemble the suction and discharge valves, leaving off the discharge valve cover. Work the fuel hand pump until all air bubbles have disappeared. Replace the discharge valve cover and build up pressure in the fuel system by the use of the hand pump.

If any difficulty is experienced in building up pressure by this means, the air has not been completely dislodged.

To Test Fuel Valves and Relief Valves

It is advisable to carry an extra length of fuel tubing with unions at both ends, to test out the fuel valves after regrinding.

Couple one end of the tubing to the accumulator and the other end to the fuel valve. Then work the hand pump to build up 4000 lb. pressure in valve.

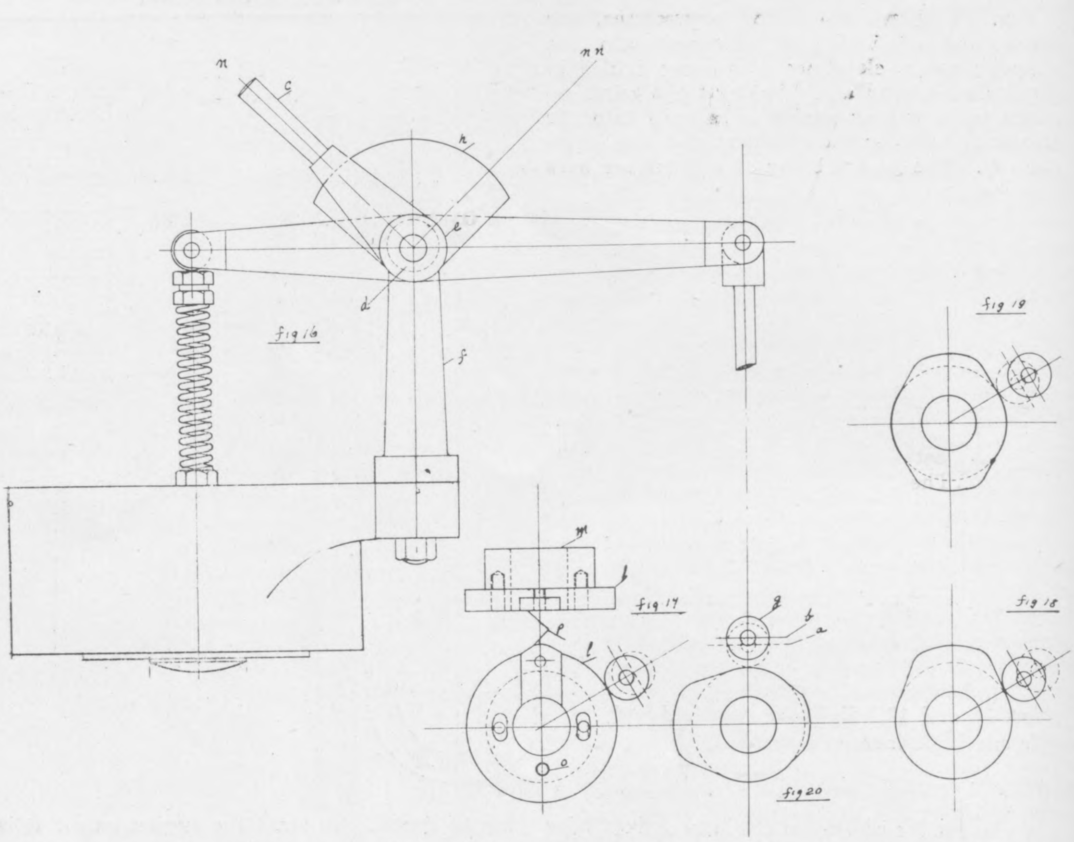
An attachment can be made to hold the valve rigid, while the needle valve is lifted off its seat to observe the nozzle spray.

The cylinder relief valve can be tested in like manner with the fuel oil hand pump. Regulate the spring tension until the relief valve lifts at the correct pressure and oil is delivered through the escape holes. Clean off all fuel oil before replacing on engine.

Remarks on Operation

Don't experiment with fuel cams by seeking to advance or retard them. These cams are set by the manufacturers to give the best results at all speeds and powers.

Don't forget to keep your air starting tanks at full pressure, and avoid all air losses by leaky joints, valves and unions. Keep your tanks filled by the engine compressor, and even if you do not have occasion to use your auxiliary compressor have it ready for instant service at all times.



Figs. 16, 17, 18, 19 *and* 20, *relating to control of airless fuel injection*

Don't neglect your cylinder relief valves. Clean and test them often. Your own safety and that of the ship depend on their correct working.

Don't forget your lubricating oil strainers. Keep them clean, and have one always ready to change over at the least sign of falling pressure on your lubricating oil gauge.

Don't forget your mechanical lubricator. Make a habit to fill it at certain times according to its capacity, and under no consideration allow it to become empty. Watch the feeds and see that each part is receiving oil.

Don't feed too much oil into the air compressor. Six drops per minute are sufficient, and it is dangerous to have your air charged with oil. If the discharge pipe from the air compressor is not water cooled the temperature of oil charged air may reach a dangerous point.

Don't attempt to work the fuel hand pump when engine is running.

Don't allow a quantity of fuel oil to enter the combustion chamber when testing the fuel valves. Remember the top of the piston is dished and will hold oil. Combustion taking place on a lodgment of oil there when starting air is admitted is dangerous.

Motortug Conway Bros., 220 hp., owned by Stanwood Towing Co., Staten Island, N. Y.

Measuring Shaft Horsepower in the Ship

Description of a Commercial Torsionmeter that Measures the Power Delivered to the Propeller Shaft

ONE of the modern methods of measuring the power developed by an engine is to determine the torsion of the shaft transmitting the power, because that factor, in conjunction with the ascertained rigidity of the material of the shaft and with the revolutions, enables a direct calculation to be made of the power which the engine is delivering to the shaft, i.e., s.hp.

In the case of oil engines the torque varies considerably during a revolution. A mean value of the torsion of the shaft must therefore be obtained. Several instruments are commercially produced for this purpose. One of them is here described and illustrated.

Indications of the apparatus are obtained by electrical means, the necessary current being taken from the ship's lighting circuit. With the method employed the results are unaffected by variations that may happen to take place in the voltage of the ship's mains or be caused by dirty or indifferent electrical contacts, which lead sometimes to the failure of electrical measuring apparatus on board ship.

The value of the torsion is indicated by the reading of a rotatable divided drum against a fixed pointer, and this figure multiplied by the r.p.m. and divided by a constant gives the horsepower transmitted by the shaft at that moment.

Two split rings are clamped on the shaft with a spacing of two or three shaft diameters between them. One of the rings is narrow and the other extended in the form of a sleeve towards it, the free end of the sleeve ring being supported on the narrow ring in order to maintain it concentric. Both the narrow ring and the free end of the sleeve ring are protected with flanges.

When the shaft is under torsion the relative movement of the transverse plane through the shaft where the sleeve ring is clamped to it is transmitted by the sleeve to the plane of the narrow ring, and the flanges move relatively, in proportion to the movement between the two transverse planes through the shaft. This movement is utilized to alter air gaps in the magnetic paths of a system of differential transformers, and the alteration to the air gaps is measured by electrical means.

The transformers consist of two laminated iron cores provided with primary and secondary windings, and they are fixed radially on the flange of the sleeve ring. The magnetic path of each transformer is completed through two air gaps and a common laminated yoke fixed to the flange of the narrow ring. When the shaft is not under torsion the air gaps of one transformer are equal to those of the other, but when the shaft is twisted one set of air gaps is increased and the other correspondingly diminished.

Primary windings of the cores are connected in series and supplied with an intermittent d.c. current, whilst the secondary windings are connected in opposition to each other. When, therefore, the air gaps of the two transformers are equal, the e.m.f. of the secondary windings are equal and opposite, and no current flows. When the shaft is twisted the air gaps become unequal and consequently the e.m.f. of one of the secondary windings becomes greater than that of the other, and a current will flow to an extent dependent on the amount of difference of the air gaps. Provided the excitation of the primary windings is constant, a measure of the current is a measure of the alteration to the air gaps and consequently of the torsion of the shaft.

There are, however, many practical difficulties in so measuring the current. The voltage of the ship's mains from which the primary current is obtained cannot be relied upon to

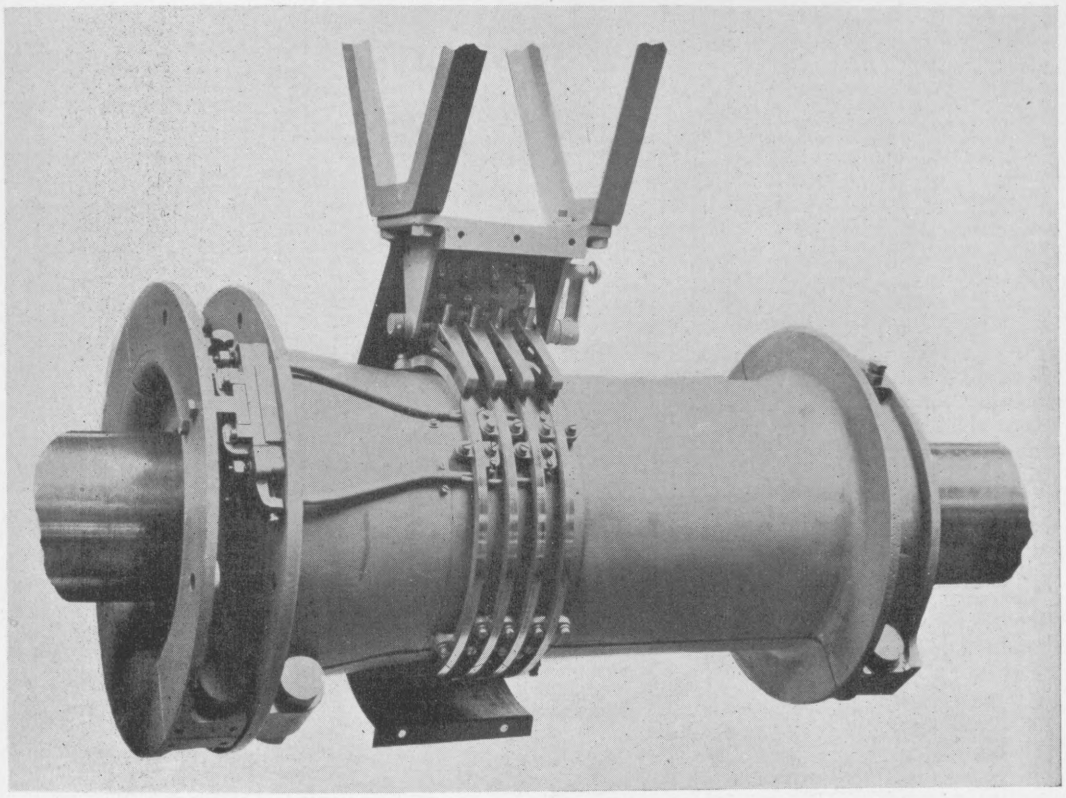


Fig. 1. General view of the torsionmeter fitted on the propeller shaft

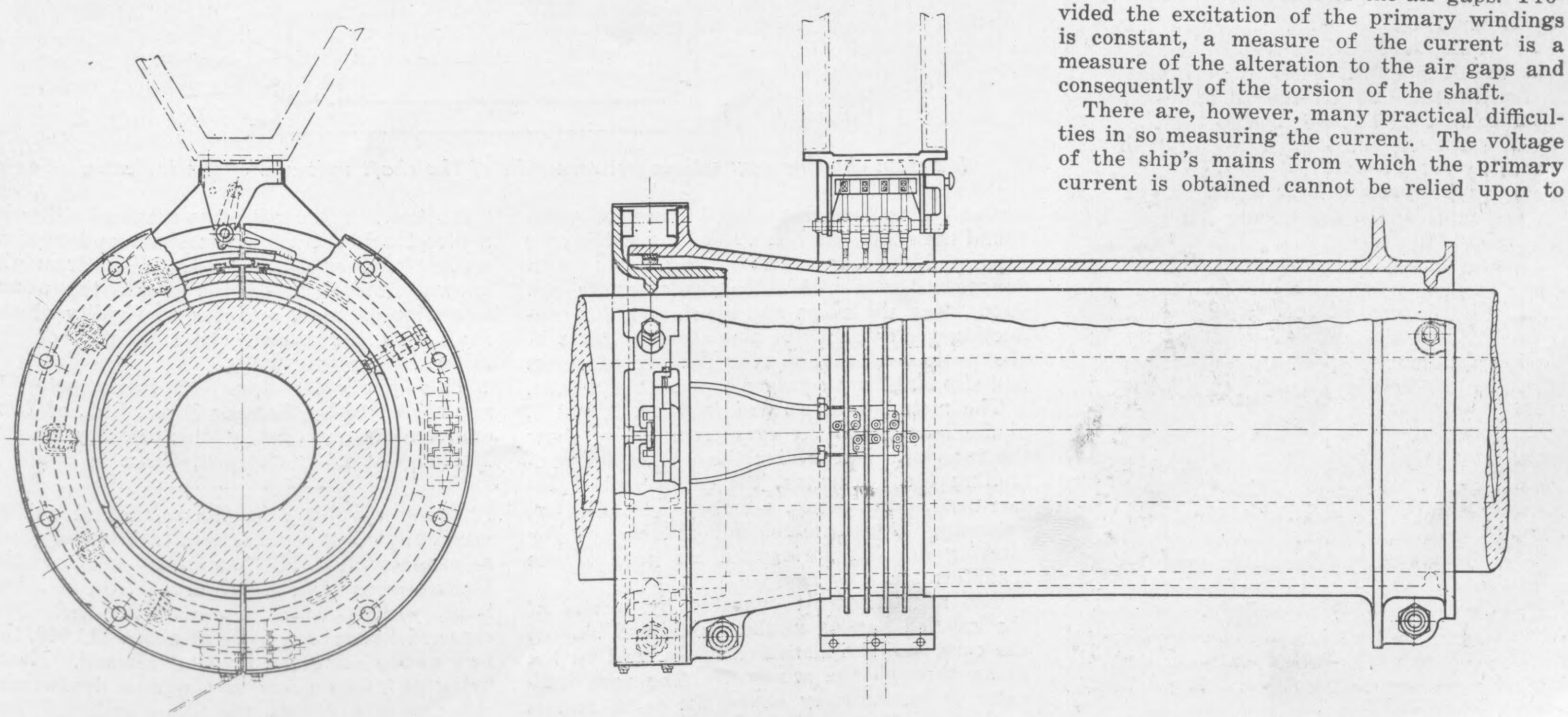


Fig. 2. End view and side elevation (both in part section) of a commercial torsionmeter for measuring power transmitted to the propeller

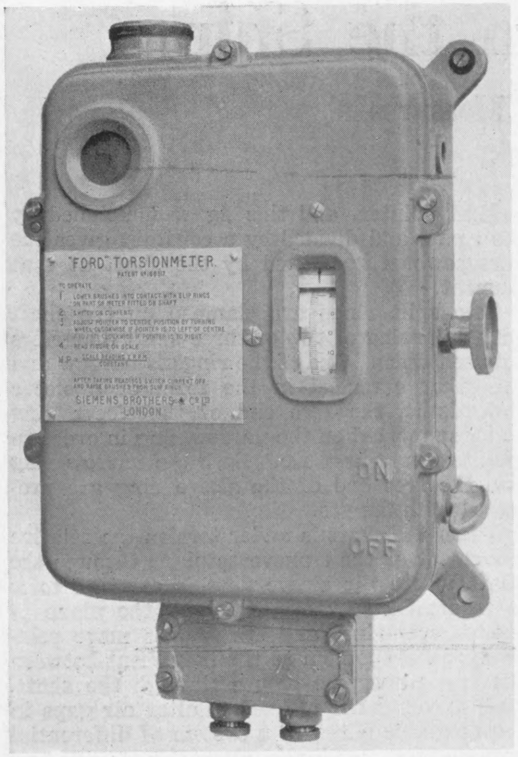


Fig. 3. Indicator of torsionmeter

remain constant, and the resistance of the contacts between the brushes and slip rings required for taking the current in and out of the transformers mounted on the revolving shaft is likely to vary and cause an alteration to the secondary current. The results are also dependent on the correct calibration of the measuring instrument.

To obviate these objections, the secondary current is not directly measured, but is opposed by a second system of transformers exactly similar to those mounted on the shaft and in which the air gaps are alterable by hand. When the air gaps of the second or indicating transformers are so altered that an equal e.m.f. is opposed to the e.m.f. of the first or shaft transformers the alteration to the air gaps of the second system is measured and a reading proportional to the torsion of the shaft thus obtained.

In the meter the H-shaped iron piece between the cores of the transformers is moved

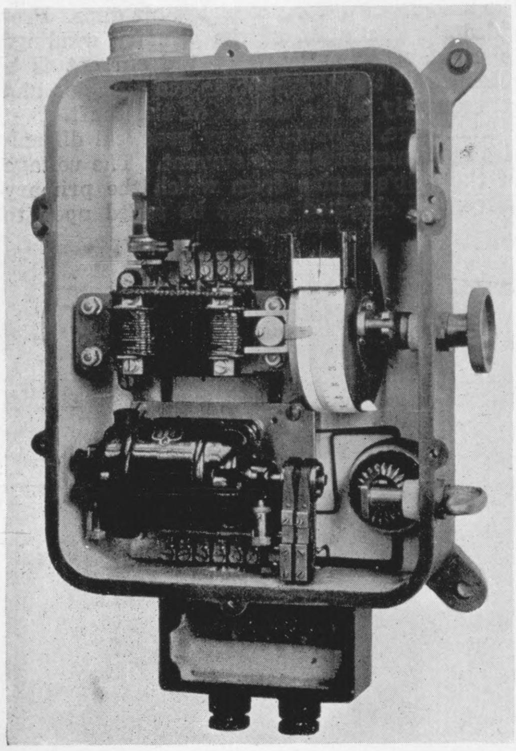


Fig. 4. Indicator with cover removed

by a micrometer screw provided with a large divided drum for indicating the movement. The "no current" or balanced condition of the two transformers is shown by a center zero indicator in the circuit.

The intermittent d.c. current for exciting the primary windings is obtained by passing the current through an interrupter actuated by a small motor, and the resultant a.c. secondary current is interrupted by the same motor to cut out one phase, so that a d.c. moving coil indicator can be used for finding the "no current" or balanced condition. To prevent rapid oscillation of the moving coil indicator electrical damping is introduced, and consequently the indication is proportional to the mean value of the fluctuating torque during each revolution of the engine.

Fig. 1 shows the portion of the apparatus mounted on the shaft and Fig. 2 shows it in

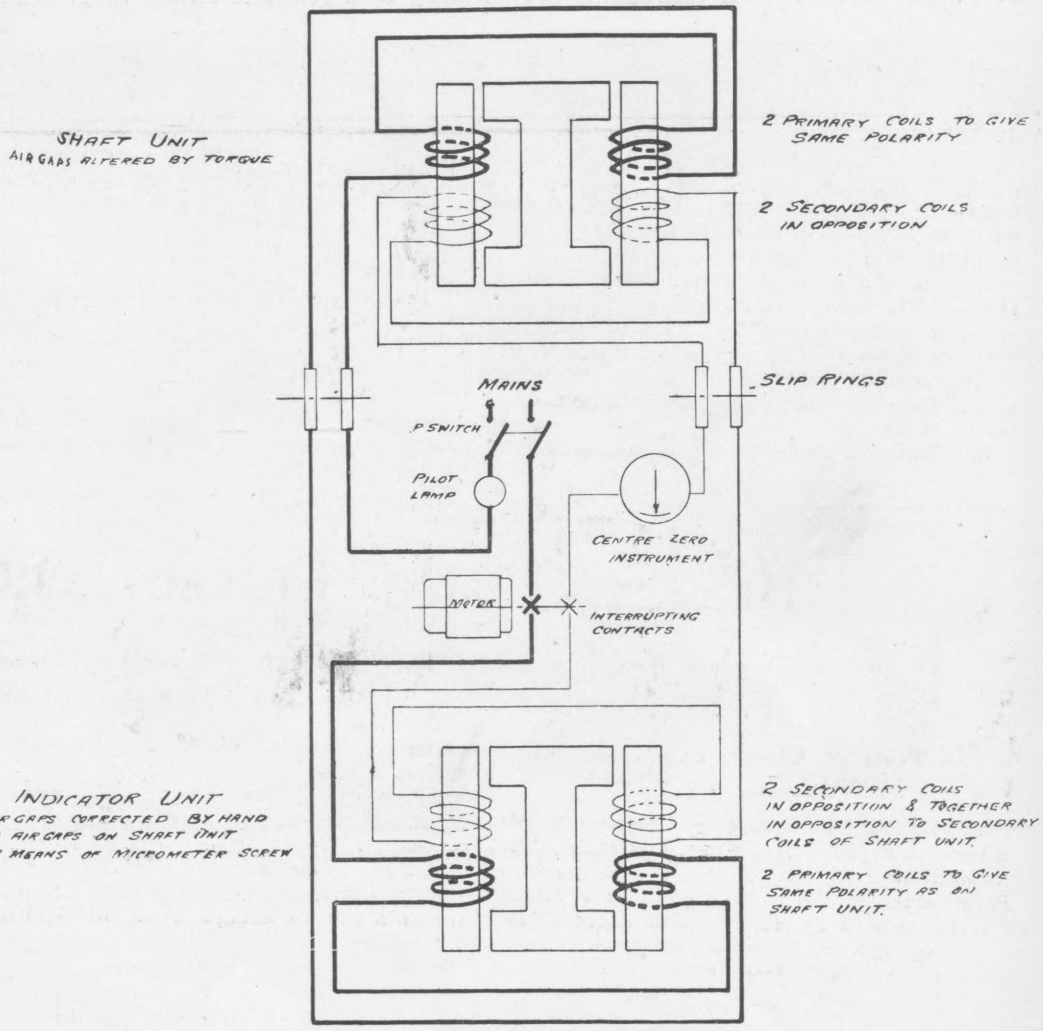


Diagram showing connections between coils of the shaft unit and of the indicator unit

section. The two rings can be seen clamped round the shaft, the free end of the sleeve ring supported on the narrow ring, both with flanges between which the transformers are fixed. The slip rings and brush gear for conducting the current are also shown. Parts of the protecting covers over the transformers and slip rings are removed to show the details.

The meter is illustrated in Figs. 3 and 4, in the latter with the cover removed to show the interior. It contains the switch for controlling the apparatus, the motor driven interrupter, pilot lamp, moving coil indicator, balancing transformers and micrometer for altering and measuring the air gaps of the transformers.

The micrometer is operated by means of the knurled handle on the right hand side of the case, and the measurement is read on the drum through the window. The zero indicator is immediately above the drum and is seen through the same window. The pilot lamp, indicating when the current to the primary coils is flowing, flashes through the circular window at the top left hand corner of the case.

Means are provided in the mounting of the transformers on the shaft to enable the air gaps to be adjusted to the "no torsion" condition of the shaft after the meter is mounted.

When the apparatus is not in use the brushes are lifted from the slip rings and the current cut off by the switch lever at the right hand bottom corner of the meter.

When one desires to take readings, the brushes are first lowered on the slip rings and the current switched on. The handwheel operating the micrometer drum is then turned until the indicator pointer is at the zero position. The figure on the drum appearing against the fixed pointer is then read, and the shaft horsepower is equal to this figure multiplied by the r.p.m. of the shaft and divided by a constant. The constant is obtained either by a direct calibration of the shaft and torsionmeter together or by calculation from the known dimensions of the shaft, the ascertained or assumed modulus of rigidity of the shaft and the known characteristics of the meter.

This particular type of torsionmeter is manufactured by Siemens Bros., Ltd., of London, and is supplied in this country by the McNab Co., of Bridgeport, Conn.

BUCKAU, the little auxiliary engined sailing vessel in which the Flettner rotor was given a commercial trial has been sold by the Hanseatische Motorschiffahrts A. G., of Hamburg, which owned and operated her. She changed hands at a price of about $10,000, the new owner being the inventor himself. Measuring 497 tons gross and with a deadweight capacity of 612 tons, she was a technical, but not a financial, success, and the former owning company has gone into liquidation.

Recent Technical Reports and Addresses

A Review of the Principal Monographs on Motorships, Marine Oil Engines and Associated Subjects

Cast Iron for Diesel Engines

By A. Campion, F. I. C. Published by the North East Coast Institution of Engineers and Shipbuilders, Newcastle, England.

Cast iron is so largely used in the construction of oil engines and the successful development of the large marine motor is so closely associated with the ability of the metallurgist and founder to produce castings for the components, especially the cylinder liner, cover and piston, that a study like Mr. Campion has made of the effect of oil engine temperatures and stresses upon the strength, growth, hardness and structure of cast iron is a welcome contribution to the art. His paper is illustrated with micro-photographs and curves.

From the layman's point of view it is interesting to note that cast iron is a rather complex mixture of iron and carbon compounds. Part of the carbon is chemically combined with the iron to form substances differing radically from either of the two constituents, and the remainder occurs as pure carbon in the form of graphite, flakes and plates. The latter are embedded in a matrix of what is really steel. In fact the matrix contains around 70 to 90 per cent of chemically combined carbon, and if it could be separated from the mass would have all the properties of steel.

Mr. Campion pleads for adequate treatment of the steel matrix as a means of securing the best Diesel engine castings. Noting that 0.90 is the carbon percentage which gives ordinary steel its greatest strength, he recommends that this be a guide for determining how much carbon the steel matrix should contain. He finds that 0.70 to 0.80 per cent actually gives the greatest strength, because at this figure a finer grain structure is obtained than at the 0.90 per cent which holds good for plain steel.

Considerations of this kind lead Mr. Campion to recommend 2.8 per cent as the maximum total carbon suitable for a Diesel engine iron. He claims that although this rather low carbon percentage tends to increase foundry difficulties by shortening the range of fluid temperature, adequate cupola or furnace control can overcome this. The advantages he ascribes to the higher-tensile iron produced are claimed to outweigh the foundry difficulties. The paper deals also with annealing and the loss of tensile strength at higher temperatures.

Electric Propulsion for Ships

By Eskil Berg. Published by the North East Coast Institution of Engineers and Shipbuilders, Newcastle, England.

In this exposition of the electric drive, the author makes the point that in the turbo-electric system there is a considerable gain due to increased turbine efficiency when compared with the geared turbine drive. On the other hand when the Diesel electric system is employed there is no increased Diesel efficiency, and we may add that, compared with a slow-speed direct drive, there is a decreased Diesel efficiency if the generator be driven by a high speed engine. This is the fundamental reason why the demonstrated efficiency of the electric drive for large warships has no bearing upon the problem of the efficiency of the electric drive for large motorships. The advantages of the electric drive in motorvessels rest upon other factors, which have been frequently set out in this magazine, and Mr. Berg has missed them altogether. In fact his acquaintance with Diesel engine performance appears to be very meager. He makes the statement that "with the advent of the adoption of high steam pressure, superheat, steam extraction for feed heating, as well as possible reheating between stages, electric propulsion makes the simplest and most ideal installation, and results can be obtained equal to, or better than, the best Diesel when total fuel consumption of the whole ship is taken into consideration." It is fair to ask Mr. Berg what he thinks is the best total fuel consumption recorded on a motorship, but it is doubtful he will answer because it will knock the bottom out of his assertion.

Review of the Latest Technical Books

Recent Domestic and Foreign Publications Relating to Ships and Engines and Their Operations

The Ports of Charleston, S. C., and Wilmington, N. C.

Prepared by the Board of Engineers for Rivers and Harbors, War Department, in coöperation with the U. S. Shipping Board Port Series No. 9. 160 pages and 9 folded inserts. 9 in. x 5¾ in. Price 50 cents. Published by Superintendent of Documents, Government Printing Office, Washington, D. C.

With the industrial development of the South the two ports covered by this volume are making steady progress. According to the latest figures (MOTORSHIP, Nov. 1925, p. 810) Charleston has an annual waterborne commerce exceeding 3,000,000 tons and Wilmington has passed the 1,000,000 tons mark, representing in the case of the former three times the volume, and in the case of the latter 1¼ times the volume of 1914.

Charleston is one of the most important fertilizer manufacturing centers in the United States, and this industry has utilized the port extensively for receiving raw products. During the last few years the port has increased its receipts of petroleum, and these two commodities now constitute about 95 per cent of the inbound traffic. The construction of the Southern Railway coal terminal in 1915 opened the way for handling a large outbound coal traffic and made unnecessary the further extensive importation of coal. Under a progressive local organization, the business of the port is developing and extending rapidly. The city is one of the few in the United States which not only owns but operates its terminal facilities with belt line communication.

Wilmington, like Charleston and other south Atlantic ports, owes its water commerce largely to the railroads which have established terminals there. It is used by one of the leading American cotton exporting houses for shipment of export cotton originating in the immediate hinterland. On a tonnage basis the exports are of less importance than the imports. The latter include many full cargoes of fertilizer materials for use in the fertilizer plants located on the Cape Fear River and Northeast Cape Fear River. The port receives, via coastline lines, large quantities of miscellaneous merchandise for consumption in the adjacent territory.

Full information about the port and harbor conditions and facilities for shipping and commerce are set out, as in the previous volumes of the series. This is the 13th volume issued, though its index number is 9 in the series.

The Ports of Savannah & Brunswick, Ga.

Prepared by the Board of Engineers for Rivers and Harbors, War Department, in coöperation with the U. S. Shipping Board. Port Series No. 10. 200 pages and 10 folded inserts. 9 in. x 5¾ in. Price 75 cents. Published by Superintendent of Documents, Government Printing Office, Washington, D. C.

Through the two Georgia ports are shipped a large portion of the so-called "naval stores" produced in the South. Quantities of lumber and a big tonnage of cotton and cottonseed products are also shipped coastwise and for export. Savannah's total waterborne commerce amounted in 1924 to about 3,400,000 tons, but Brunswick's is only about one-fifth of that figure.

This volume, like all the others of the series, 14 of which have been issued, contains full information regarding the port and harbor conditions, port customs and regulations, services and charges, fuel and supplies, piers, drydocks, repair plants, warehouses, railroad and steamship connection and terminal charges.

In Savannah the principal piers and wharves are owned and operated by the Seaboard Air Line, the Central of Georgia and the Atlantic Coast Line railroads. The Central of Georgia Railway has a subsidiary, the Ocean Steamship Co. maintaining a service to New York and Boston and it leases a terminal to the Merchants & Miners' Transportation Co., which operates a coastwise service between Jacksonville, Savannah, Baltimore and Philadelphia. The port occupies an important position in the coastwise trade and handles water-rail traffic to and from points as far West as Salt Lake City.

Increase of cotton and cottonseed manufactures in the Southern States has helped to decrease the shipments of these raw materials out of Savannah very considerably during the past 10 years, but the establishment of a local sugar refinery requiring about 150,000 tons of sugar per annum has offset the drop in exports by a gain in imports.

Brunswick, at the mouth of the Altamaha River, is the main tidewater terminus of the

(*Continued on page* 304)

Sketches and Working of Oil Engines*

Valve-Gear Fundamentals Illustrated by Diagram and by Views Showing Effects of Wrong Operation

THE valve gear of any engine is its nervous system. Like that of the human organism, it translates minute stimuli into action and power, registering both external controlling influences and the reflexes produced by the motion of the engine parts among themselves.

The valve gear of an oil engine fits into this parallel better than does that of a steam engine. Whereas a slide valve has nothing to do with the liberation of energy from its ultimate source (the fuel) the valves of an oil engine are, on the contrary, intimately concerned in this most vital function. A steam engine valve is hardly more than a barrier, periodically removed and replaced, between the boiler and the cylinder. There is nothing in the steam chest, for instance, to affect the maximum pressure to which the engine is exposed, the pressure being subject to an entirely different set of influences often originating many yards distant from the actual engine mechanism.

A steam engine valve unites in itself the two extremely simple functions of admitting and exhausting vapor, and this vapor is a distilled product with a degree of purity that is exceptional so far as the ordinary run of fluids handled in mechanical engineering work is concerned. It has an exceedingly moderate temperature and, barring the most abnormal derangements, is physically incapable of conveying solids or other injurious materials into the engine.

Oil engine valves, on the other hand, are directly exposed to the fire of combustion. Because they are small they heat up readily, and on account of their motion cannot easily be relieved by water cooling. Fuel and air entering an oil engine carry with them quantities of solids, and the combustion process converts some of the fuel constituents into crystalline substances and others into carbon and coke.

Painting the picture thus blackly might encourage the belief that oil engine valves are unduly troublesome, just as the high cylinder pressures discussed in previous chapters might suggest that engine frames might be hard to build strongly enough. But the extra demand on the valve designer's skill has stimulated him to higher efforts, which may fairly be said to have more than overcome the difficulties. That is why reports come in about marine Diesel engines making non-stop runs of 50 days or more at full power. The valves of the double-acting engines on the motorliner GRIPSHOLM, for example, are 12¾ in. diameter over the heads and sustain a maximum load of 32 tons at a maximum temperature running to furnace heat. Yet all 24 of them maintain tight joints on a seat length of 40 in. each throughout the duration of a transatlantic voyage with a mean effective indicated pressure exceeding 95 lb. per sq. in. The chief engineer of the vessel expresses himself ready to turn round and go back again across the Atlantic at 20 minutes' notice. Such demonstrations may be regarded as a measure of the skill that tempers the severe conditions whereunder oil engine valve gears are operated.

There is a fundamental difference of principle also underlying the operation of the sliding type of valve used in steam work and the poppet valve characteristic of the internal combustion engine. For some reason the slide valve idea appeals to the imaginations of inventors far more than does the poppet valve principle, with the result that floods of patents relating to rotary and sliding valves on internal combustion engines threaten to swamp the Patent Office each year. With so much talent brought constantly to bear on this problem it is surprising that little has been accomplished thus far beyond the successful application of the Knight sleeve valves, but, whereas this development seems to be maintaining itself commercially, there seems to be no reason for expecting that it will become engrafted on established practice.

The clear and sharp demarcation between slide and poppet valves is that the slide valve remains continuously in contact with its joint face, whereas the contact between a poppet valve and its seat is periodically interrupted.

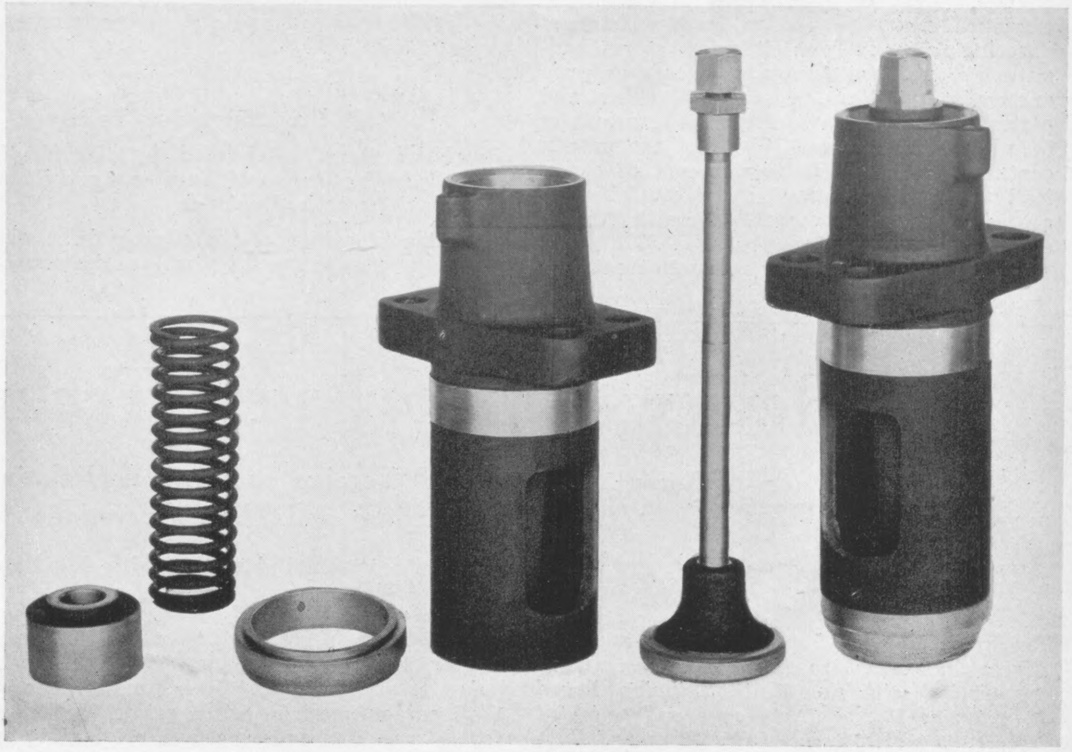


Fig. 131. *Exhaust valve parts and assembly showing the characteristic poppet type*

This fundamental distinction leads to a number of other almost equally vital differences.

There is no friction between a poppet valve and its seat. No lubrication is therefore required, and the destructive influence of solids is noticeably less than on rubbing valve surfaces. Steam engine slide valves already begin to suffer from lubrication difficulties as soon as superheated steam at a few hundred degrees F. begins to be used with them; what may they then be expected to do while passing hot gritty gases with temperatures running up beyond the thousand? The mere fact that flat metal surfaces of this kind, even when unbroken by ports, exclude the possibility of film lubrication, is a serious handicap and requires the intensity of bearing pressure between the two surfaces to be carefully limited.

As far as gastightness is concerned, high metal pressures are an advantage, and the balancing devices sometimes used on steam slide valves (fig. 132, *h*) are intended to leave the maximum unbalanced seating force that will not too quickly ruin the valve faces. Needless to say, the force which would be required to make a slide valve tight against the pressures encountered in Diesel engine work would also score and cut the faces beyond all possibility of use. There are no conditions commonly met with in practice to limit the intensity of bearing pressure between a poppet valve and its seat. Those minute inaccuracies which remain after machining and careful grinding can readily be crushed down by the forces with which it is usual to operate such valves. At the same time the absence of ports and bridges in the joint surface permits the closing pressure to be concentrated in a narrow circular ring with an extent of area so limited as to give high bearing intensities with relatively moderate total forces applied to the valve stem.

External mechanism for operating poppet valves differs from conventional steam engine gear for two basic reasons. Periodicity of events is produced by the slide valve with the registry and non-registry of ports while the motion of the valve itself is substantially uninterrupted. The poppet valve causes closing and opening events by contact and separation and remains motionless during considerable periods of time. As the slide valve is provided with overtravel and moves parallel to its own joint faces, it may be moved by positive and rigid mechanisms. In the poppet valve a rigid and unyielding stop in the form of an oblique valve seat positively limits the travel of the poppet and therefore makes the use of a positive mechanical drive impossible in the great majority of poppet valve applications. Although some positive drives are found in practice, interest in them is confined largely to scientific circles.

Reference to fig. 132 illustrates diagrammatically some of the points discussed above. At *A* is shown a conventional slide valve driven by an eccentric. As it is assumed always to be tight against the face *f*, its lateral position affects only the timing and sequence of events. Accidental influences of a minor character, such as the running clearance *c* on the eccentric strap therefore produce no leakage. The slight displacement *a′* correspond-

* Summary of a Course of Instruction at the Polytechnic Institute of Brooklyn, N. Y., by Julius Kuttner, B.Sc., Licensed Chief Engineer, Editor of OIL ENGINE POWER and Associate Editor of MOTORSHIP. This is the thirteenth chapter, the first one having appeared in the January, 1925, issue.

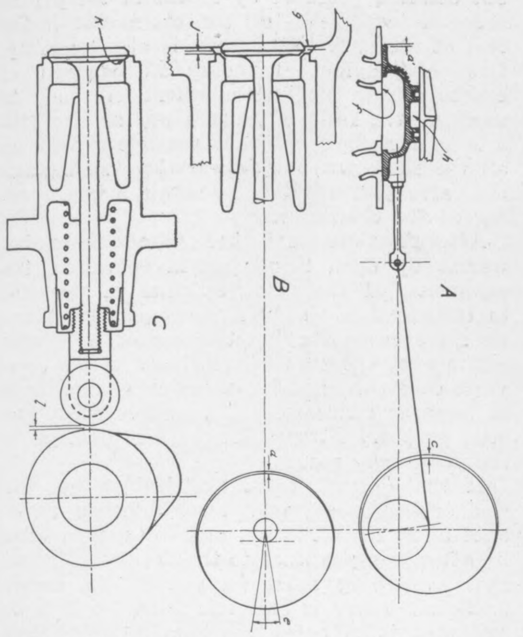


Fig. 132. *Poppet and slide valve compared*

ing to the slack in the eccentric has no noticeable effect on the operation of the valve. At *h* is shown the balancing device to prevent the destruction of the valve surfaces by excessive metal pressure at *f*.

B, fig. 132, is intended to represent schematically a poppet valve driven by any positive mechanical means in which rigid mechanical elements pull the valve away from its seat and push it back. Slack *d* in the drive produces a corresponding displacement *j* at the valve seat. This is a leak, not merely a negligible inaccuracy as at *a'*. It would permit the continuous escape of fire across the seat of the valve and would doom it to quick destruction. Supposing that contact could actually be secured at *j*, it is obvious that the amount of force exerted would be subject to a variety of accidentals such as differences in expansion due to temperature. A slight miscalculation in setting the effective length of the driving mechanical train would perhaps result in its being ruptured. At *e* is indicated the reason why an ordinary eccentric cannot be so arranged as to produce a proper sequence of valve events: the valve would remain shut, if at all, only over a vanishingly small period of time.

Many considerations of this kind have led to the almost universal adoption of the cam-and-roller drive sketched in its elements at *C*, fig. 132. It is distinguished from the eccentric motion by two fundamentals, one of which is that it leaves the choice of valve events unrestricted and the other is that it requires the use of a spring for the return phase of the motion produced. As with the eccentric, clearance *l* must be provided somewhere in the mechanical train in order that it may be mechanically workable. Because a spring is used, however, the slack *l* can never manifest itself at *k*, the seat, and the production of leakage from this cause is therefore ruled out. The spring therefore performs two functions of pivotal importance, one of these being the maintenance of the right joint surface pressure under all conditions and the other being the completion of the return phase of the cam motion.

The apparently overwhelming evidence which it thus seems possible to amass in favor of the poppet valve actuated by a cam-and-spring mechanism may some day attract the notice of rotary and slide valve inventors. Some day also, it might be demonstrated in just what respect the slide valve is superior, assuming that it would be possible to devise one capable of operating under oil engine conditions.

Watching the operation of a cam-driven valve suggests to the new observer that its motion is nothing more than a jerk. Actually, however, all cams produce clearly-defined motions capable of being subdivided and studied in their individual phases. In most cases the opening and closing motions of a cam-operated valve are reversed duplicates.

In its travel from the closed to the open position the valve is brought quite gradually from zero speed up to a maximum velocity. As the opening movement continues to progress, the highest speed begins to be reduced again, in much the same way as a locomotive engineer closes his throttle and applies brakes while approaching a station. All of this happens between the time when the valve is disturbed from rest in its closed position until its motion ceases again in the wide-open position. The acceleration and retardation occurring here are of course not to be confused with backward and forward movement comprising the entire action of the cam.

According to the fundamentals of mechanics, force must be applied to the cam roller in order to change its condition of rest into that of motion at a definite rate of speed. The more sudden the change, the greater must be the force exerted by the cam. Note that this force is due solely to the change of motion and would arise even if there were no spring to oppose the motion of the roller. As a matter of fact, the spring force is unnecessary as long as the cam is increasing the speed of the roller away from the camshaft centerline; at that time the roller would stay in contact with the cam face merely by virtue of its own inertia.

surface, so the decrease in velocity requires the reversal of the push and the application to the roller of a force which is opposed to the direction of its travel and which "kills" its speed. If it is to have a period of rest in the open position beginning at the moment when the point *f* on the cam touches it, its speed must also be zero at that point.

Now, as it is obviously impossible for the cam to exert any tension on the roller, the reversed force causing retardation must originate somewhere else. The source of this inward thrust is the valve spring communicated through the tappet on the end of the valve stem and through the levers, push-rods, and whatever else may be interposed. Were it not for this restraining force the roller would fly off the cam at the point *f* and would continue with the velocity which it had there until brought to rest as the result of friction or of mechanical interference. The spring, it will be recalled, has been producing pressure between the roller and cam throughout the previous phase of the opening motion and has been added to the inertia force noted in connection therewith.

After the point of *e* has been reached and passed, the spring force is utilized to retard the roller at a rate determined by that portion of the cam outline situated between *e* and *f*. During this phase the amount of available spring force is greater than that actually necessary for producing the retardation and the excess is represented by the force of contact between the roller surface and the cam face. Note that in this case the inertia force

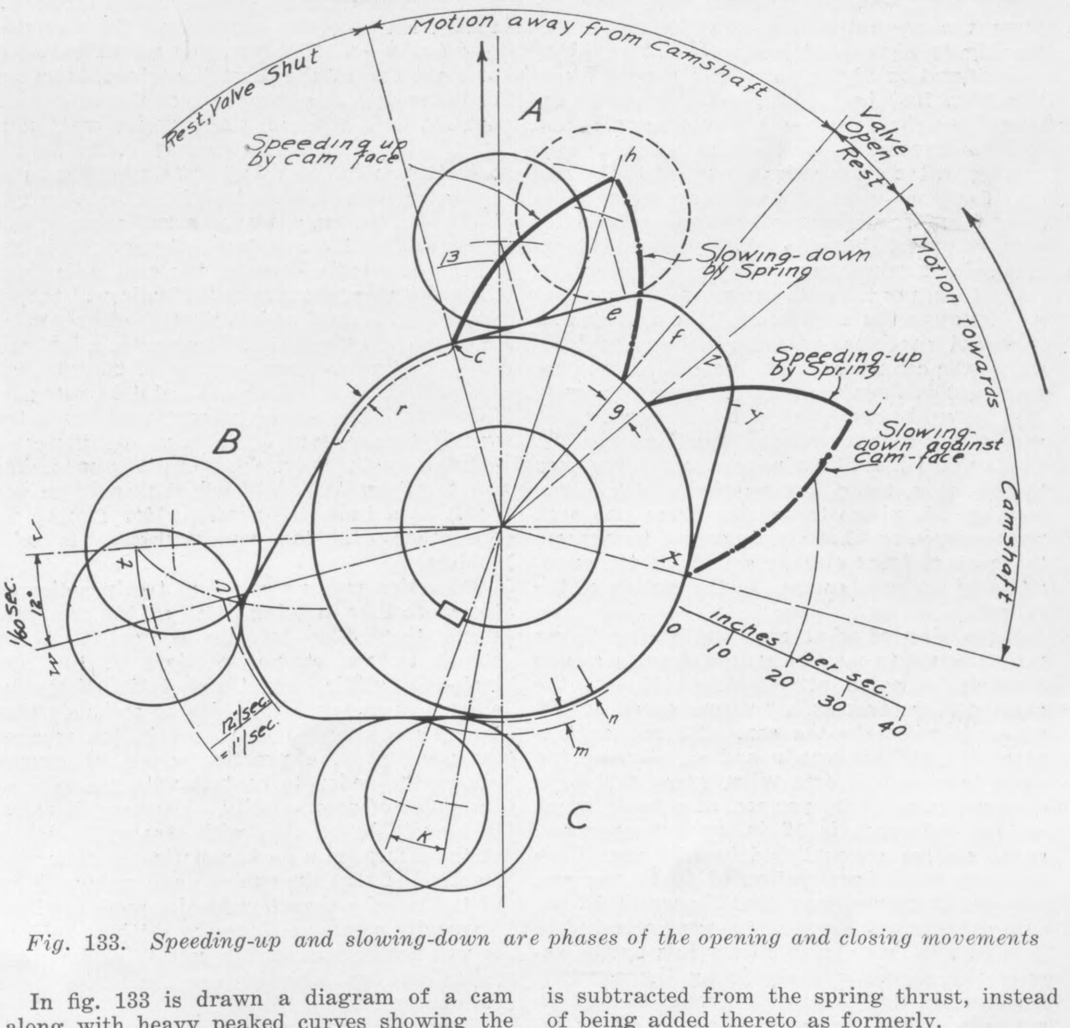


Fig. 133. *Speeding-up and slowing-down are phases of the opening and closing movements*

In fig. 133 is drawn a diagram of a cam along with heavy peaked curves showing the cam-roller speeds produced by it. Between the contact points *c* and *e* the roller is being speeded up in a direction away from the center as discussed above. Between *e* and *f* the outward motion still continues, but is accompanied by the marked drop in rate of travel indicated by the dot-and-dash curve falling off from the point *h*. Just as the increase in speed along *c*—*e* requires a push on the roller exerted in an outward direction by the cam is subtracted from the spring thrust, instead of being added thereto as formerly.

During the period of rest on the "nose" of the cam, which is concentric with the axis of rotation, the static spring force is exerted on the cam face without suffering any addition or subtraction due to inertia effects. It is the designer's aim to make the period of rest as long as possible in the interest of giving the valve the greatest possible capacity for passing gas between opening and closing times. As the latter are fixed in accordance

with the travel of the engine piston, it is plain that lengthening the rest period decreases the amount of time available for jerking the valve open and for returning it to its seat. The more suddenly this is done, the greater are the forces to be exerted by the cam and spring. Given a definite percentage of "wide-open" time, high-speed engines require greater valve forces than low-speed machines. Greater forces are therefore involved, and they explain the reputation for noisy operation belonging to high-speed engines.

Acceleration of the valve in a closing direction (fig. 133) begins at z and extends as far as y. As the roller now moves inward, and as it is impossible for the cam surface to be pulling it, the spring alone must be the source of its action. Here again the spring supplies more force than is necessary to speed the roller up to the peak j of the velocity curve, the excess being again taken up by contact pressure. At y, then, the roller has acquired its greatest speed under the influence of the spring and it would continue inward at that speed towards the camshaft center were it not for the fact that it finds the cam outline in its way. The latter, however, is constantly receding at such a rate that the roller following it has lost all its speed just at the moment when it comes to rest at x, the closed position of the valve. During this phase the spring force has been unnecessary, and has simply served to increase the contact force already set up by the opposition of the cam surface to the motion of the roller.

Some idea of what cam and spring forces may amount to may be gained from a simple example indicated at B of fig. 133. If the camshaft revolves at a uniform speed of 120 r.p.m. it will take the cam 1/60 sec. to traverse the 12° between v and w. During the same interval the cam roller speed will have changed from 16 in. per sec. at u to 28 in. at w; the difference is 12 in. or 1 ft. per sec. speed change accomplished in 1/60 sec. This amounts to an acceleration of 60 ft. per sec. per sec. Assuming now that the weight of the cam roller and the rest of the gear attached to it is 64.4 lb., we may multiply this by the acceleration figure and divide by the constant acceleration due to gravity (equal to 32.2 ft. per sec. per sec.) to get the amount of the acceleration force in pounds:

$$\frac{60 \times 64.4}{32.2} = 120 \text{ lb.}$$

This is the net accelerating force coming from the spring, the surplus produced by the latter being taken up as roller pressure on the cam face. If we assume that the cam is going the other way and that the roller is coming to rest near the top of the cam, the 120 lb. represents the force which the spring must exert in order to prevent the roller from leaving contact with the cam surface.

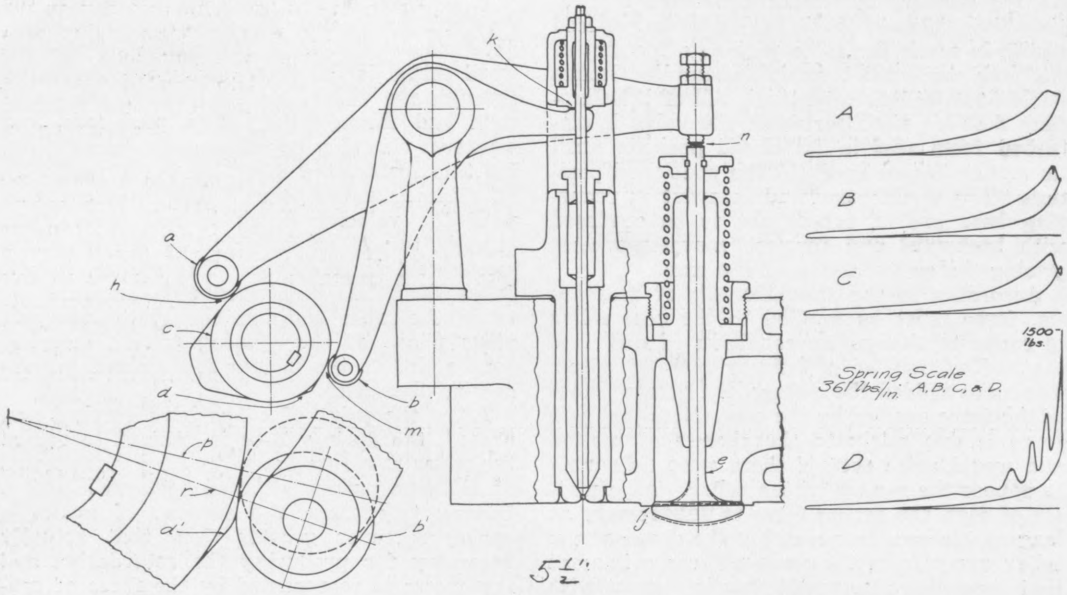


Fig. 134. Effect of cam roller clearance on operation of exhaust and fuel valves

With attention concentrated on the foregoing it might be easy to gather the impression that the spring could really be dispensed with at the rising and falling ends of the cam, assuming for the sake of argument that a suitable mechanical arrangement would throw the spring out of action except over the stretches e—f and y—z. However, that would leave out of sight one of the most important functions of the spring, this being to keep the valve hard against its seat at all times except when it is open. In order that this may really be the case there must be no misunderstanding as to the point where the spring thrust is taken up: there can be only two choices, namely, the valve seat or the cam face. Somewhere in the mechanical train between the cam roller and the valve stem there is an adjusting or tappet screw by means of which the clearance n, measured by means of "feelers" against the lower concentric cam portion o, may be adjusted. It is, of course, absolutely essential that this clearance be present to prevent the spring thrust from being taken up there instead of on the valve seat. As it is impossible for the force to be exerted at more than one point at a time, the necessity for freeing up the roller over the cam portion o is self-evident.

Objection may be raised to this practice on the score that in being lifted off the cam base circle the roller becomes subject to being struck by the steeper portions of the cam instead of rolling sweetly on to it. Some designers have taken this idea so seriously that they have provided clearance in the manner indicated at r, a practice which, of course, allows the roller to contact with the cam at the point of zero velocity. However, it seems to be established that with reasonable values of the clearance n no hammering or slamming results and that the expense and awkwardness of the relief r therefore hardly seem justified. Naturally excessive clearance settings such as m will not only cause the cam to hammer the roller, but will also derange the timing by a serious amount k.

Keeping the clearance n at a standard value is therefore of twofold importance. Indiscriminately shoving a feeler gauge between the roller and the cam will not give a satisfactory indication unless the precautions generally recommended for fine mechanical measurements involving thousandths of an inch are observed. While making the roller clearance measurements it is advisable to pry up the roller with a piece of flattened brass pipe once or twice, and then to maintain a slight, but definite, pressure by means of the pry in order to insure that all the lost motion in the rest of the mechanical train is also taken up. Lack of judgment in using the pry will of course throw the measurement off just as much as the failure to use a pry at all; with a little care, however, it is possible to take up all the slack without appreciably "springing" the valve gear and thus to obtain a true reading of the clearance.

After clearances have been set and the engine warms up upon being put in operation, the expansion of the valve spindles in response to temperature rise will again change them. In some cases the lengthening of a spindle will thrust the cam roller closer to the camshaft centerline and in others expansion tends to increase clearance. Unless this effect has been properly allowed for, serious damage to the valve may result.

If the exhaust valve sketched in fig. 134 was originally set with an insufficient clearance at h, say 0.015 in., and it expands 0.040 in. after being exposed to the fire, it is obvious that something must happen to the excess 0.025 in. There is only one thing for it to do and that is to cause the cam roller to bear hard on all parts of the cam and to cause the valve to stand open as indicated at j. As the roller is generally of glass hard steel running on a pin of similar material the two are very apt to "freeze" together in a manner that may make it necessary to scrap the entire valve lever. But a far more serious consequence is the permanent leak established at j. Not much reflection is necessary to show that fire sweeping over these metal surfaces at temperatures of thousands of degrees F. will fuse or otherwise quickly ruin them. The indicator card which is often produced by this particular form of derangement is sketched at C, fig. 134, and near the top dead center plainly shows the loop marking the rapid fall in pressure due to loss of gas through the leak. Fig. 135 shows the intense blow-torch action to which the valve is being subjected at the same time.

Decreased clearance of fuel cam rollers may result in danger to life and limb. A fuel needle continuously held open because of spindle expansion admits fuel during the compression stroke; preignition results and the pressure due to the burning of the fuel, instead of counterbalancing expansion, is now

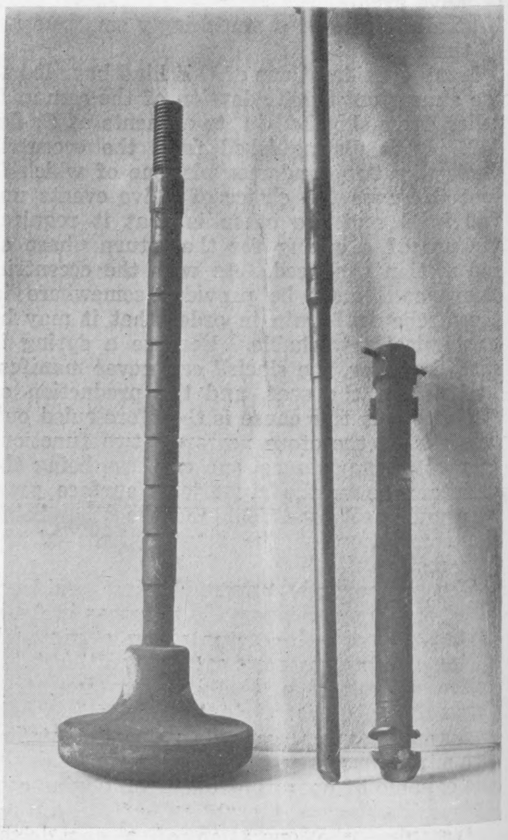


Fig. 135. Valves burnt by leakage

Fig. 136. Ruptured injection air line

added to compression. If the engine is not immediately wrecked and the injection air pressure is high enough, an indicator card similar to that shown in fig. 134, *D*, may be taken. Otherwise the excess pressure backs up into the open fuel valve and fuses together its interior parts as shown in fig. 135. Lubricating oil vapor possibly carried along with the injection air may allow an explosion to be communicated along the air tubing, converting it into ribbons as depicted in fig. 136.

Dire effects like these cannot, of course, result from the expansion of a fuel needle arranged like the one shown in fig. 134, because there the effect of expansion is to increase the cam roller clearance. However, there are any number of engine designs in which fuel needle expansion may cause exactly the same kind of a leak as that illustrated by means of the exhaust valve sketched at *e*.

But even when lengthening of the needle does not invite explosions, increase of cam roller clearance becomes highly objectionable for other reasons. Owing to the extremely gentle slope of the fuel cam nose, *d* and *d'* the added clearance makes a substantial difference in the point at which the cam face and the roller make contact. The more slack there is the longer obviously will the cam travel before beginning to lift the roller. Ignition delayed from this or any other cause makes the engine pound heavily. Fuel entering late does not catch fire immediately, but when it finally does ignite there is so much of it present that its sudden combustion produces an abrupt pressure rise. Because of the greatly lowered temperature, combustion is imperfect, is accompanied by after-burning and causes the engine to be fouled up with coke and carbon (fig. 134, B). All of these things may happen to the operator who believes he has set his clearance right while the engine was cold, but who did not trouble to ascertain whether he would have the same clearance after the engine settles down to its normal running temperature. The following table tells the story:

Exhaust Valve Clearances

Cylinder	I	II	III	IV	V	VI	
Cold	0.037	0.037	0.037	0.037	0.037	0.037	inches
Warm	0.016	0.026	0.018	0.016	0.014	0.028	"
Reduction	0.021	0.011	0.019	0.021	0.023	0.009	"

These are actual readings taken from an 800 s.hp. six-cylinder engine. Had the clearance on No. VI, for instance, been 0.02 in. the exhaust valve would have begun to leak seriously. The table shows the importance of measuring clearances quickly as soon as the engine is shut down after the first full-power run and using these readings for correcting subsequent clearance adjustments.

Only when the clearance becomes excessively great does it begin to affect the timing seriously, as already explained. Moreover, unreasonably large clearances cause the cam to strike the roller a sharp hammer blow, and if tolerated for any length of time may cause tappet screws or other parts to batter loose. Neglect of this kind has been known to proceed so far as to permit the valve spindle to drop with its valve bodily into the cylinder, breaking the piston, cracking the cylinder head and doing other serious damage (fig. 137).

Assuming that the clearances of the rollers for inlet and exhaust valves are properly looked after, they do not have any appreciable effect on the timing of their events; in this respect therefore large valve clearances differ from those used on fuel valves. Whereas in the case of a steam engine the length of the valve rod will have far-reaching consequences on the steam-lap, cut-off and the like, the cams of an oil engine are permanently keyed to the camshaft and their outline is determined at manufacture.

Besides admitting air and exhausting spent gases, the valve gear of an oil engine must also provide for the introduction of fuel into the cylinder, this being carried out by a special valve which will be made the subject of detailed consideration later. The admission event of an oil engine concerns only air and, as it has nothing to do with load regulation like the supply of steam to the steam engine, it is subject neither to special adjustments nor to governor nor throttle control. The exhaust process is naturally also fixed and invariable.

Nevertheless, the oil engine operator wishes to know about the admission of air and the exhaust of spent gases, particularly because indicator cards as ordinarily taken do not give much of a clue to these events. Most indicator cards are taken from oil engines purely for the purpose of studying fuel injection and combustion and are drawn with scales varying between 400 and 500 lb. to the inch.

The aspiration of air into a cylinder takes place perhaps at a fraction of a pound below atmosphere, whereas only a brief portion of the exhaust process occurs at more than 3 lb. per sq. in. Naturally, such pressures as these affect a 400-lb. spring hardly at all, with the result that most of the important suction and exhaust events are all but obliterated.

To facilitate the study of suction and exhaust events an indicator with a weak spring scale—about 40 lb. per inch—is frequently employed. A positive mechanical stop limits the upper end of the indicator piston and pencil motion and causes a straight horizontal line to be drawn as long as the pressure in the cylinder exceeds the pressure corresponding to the height of the line. Naturally, that eliminates about 90 per cent of the normal diagram comprising most of the compression strokes, but it brings out the suction and exhaust phases with a degree of clearness directly proportional to the ratio of the spring scales.

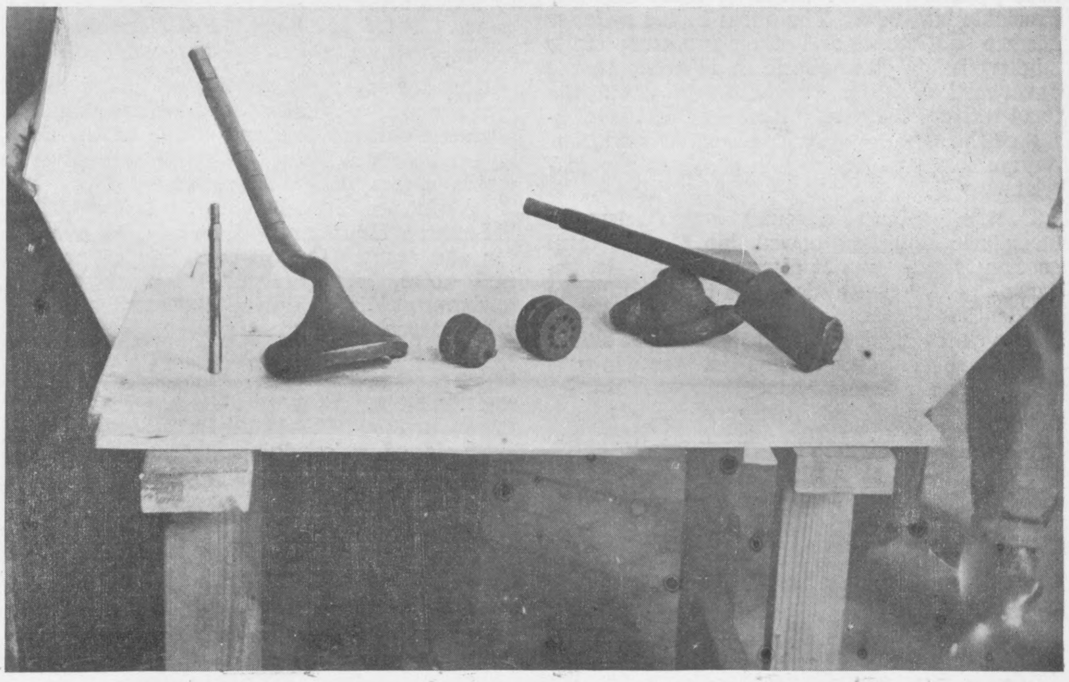


Fig. 137. Valves dropped into cylinders because of neglected adjustments

In fig. 138 is shown an actual card taken with a spring scale of 36.2 lb. per sq. in. from a 13 in. x 20 in. Diesel cylinder rated to develop 50 hp. at 210 r.p.m. After rising to a height corresponding to about 59.5 lb. per sq. in. the indicator piston rested against a stop and drew a line which, on the original card, was situated a shade less than 1⅝ in. above the atmospheric line.

That part of the curve marked *k* shows the tail-end of expansion occurring just before the exhaust valve begins to open. Owing to the sudden acceleration of the pencil motion downward a wave has been produced in this part of the line, and it therefore probably does not give a true picture of the pressure conditions within the cylinder. At *a*, however, there may be observed a definite break in the smoothness of the line which may be taken as evidence that gas is beginning to leak out past the gradually opening valve. As the effective opening of the latter begins to increase rapidly, the sharp vertical drop at the end of the diagram is produced, and with the full opening of the valve available an energetic outrush of gas begins to take place.

Acting somewhat like a solid piston, the escaping body of gas advances a little faster than the remaining portion within the cylinder can follow it, with the result that the slight vacuum indicated at *j* is formed. But it does not continue long, and at *h* may be observed the effect of the piston pushing gas before it

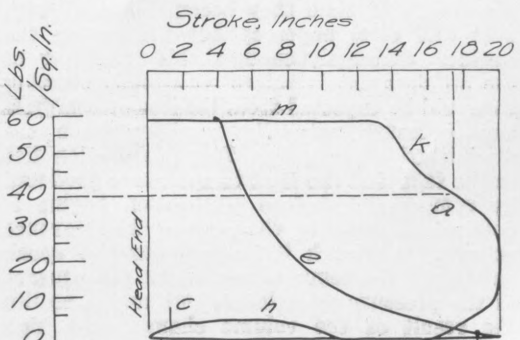


Fig. 138. Weak-spring indicator card

and slightly compressing it—to about 4 lb. per sq. in. This pressure is probably somewhat higher than it should be and may be attributed to a long line of exhaust piping. Improper muffling devices will of course produce a similar effect. In the Maxim silencer ingenious use is made of sound diffraction to dissipate the energy of the sound wave. As the latter is an oscillatory disturbance differing radically from the bodily translation of the outrushing gas charge, it can be broken up by means of its own echoes, produced in spiral chambers offering a negligible amount of resistance to the progress of the gas current. Unnecessary resistance to the outflow of exhaust gas, be it caused by insufficient valve area, improper timing or unsuitable exhaust passages beyond the engine, adversely affect its operation. About 7 per cent of the cylinder volume—what is known as the clearance or combustion space—is bound to remain full of spent gas anyway. If this is augmented by any impediment to the outflowing gas, the next incoming charge will be heated, diluted and in other ways prevented from supporting combustion at maximum efficiency.

Although the mark *c* on what is still the exhaust line indicates that the inlet valve is beginning to lift, and correctly so, a little before head-end dead center, the opening at first produced is too small to exert any appreciable influence. The gases in the cylinder become quiescent and their pressure falls slightly below atmospheric in response to the downward recession of the piston from the head-end dead center. Here again, the closing of the exhaust valve, marked *b*, is without appreciable influence on the shape of the diagram.

Suction begins at a pound or two below atmospheric along the bowed line *f*. The term suction is perhaps not altogether suitable, because it conveys the idea that the air already inside the cylinder is exerting actual tension on the air situated just outside the inlet elbow. What really happens is that the pressure with which the air inside the cylinder resists the normal atmospheric pressure outside is diminished by expansion attendant upon the increase of volume. As this drop in pressure is not entirely offset by the air being pushed in from the outside, the line *f* must be subatmospheric.

Anything that increases the resistance to the inflow of air is harmful to the engine. The less air there is present the more sluggishly the incoming charge of fuel will burn, with the result that the flame is maintained over an abnormally long period and gives up extra heat to the confining metal surfaces. If the air supply is ample, on the other hand, combustion takes place promptly and efficiently, a greater share of the heat developed is converted into mechanical work, while a correspondingly smaller proportion remains available for producing unnecessarily high metal temperatures.

Suction guards consisting mostly of slotted pipes are generally fitted to the intakes of an engine to prevent such objects as balls of waste, engineers' hats, mice's nests, and the like from being drawn into the cylinder. In the course of time they become clogged with dust and dirt, imperceptibly suffocating the engine. Strangely enough, this throttling of the air does not interfere with ready starting as would be expected from the loss of compression pressure naturally resulting from the diminished air supply. The explanation lies in the fact that the final temperature produced by compression before combustion begins is determined solely by the ratio of the clearance volume. Although it is customary to speak as though the temperature is the consequence of the pressure established, both are in reality the result of the volume change. At high altitudes Diesel engines may run with compressions 80 per cent lower than normal while the final temperature of compression is still the same as at sea level.

Hence the clogging of inlet air breathers may remain undetected and in many cases where indicator cards are taken that reveal it, the loss of compression is frequently attributed to leaky piston rings, valves, and the like. The worst remedy for the trouble is cleaning the breathers while they are in place, irrespective of whether the engine is running or not. Most of the dirt drops inside and is sooner or later drawn into the engine. There it scores liners and pistons and may even find its way into the main bearings via the force feed lubricating systems. The breathers should be periodically cleaned away from the engine as a fixed part of operating routine.

Review of Latest Technical Books

(*Continued from page* 295)

Atlanta, Birmingham & Atlantic Railway and has connections with the Southern Rwy. Co. and the Atlantic Coast Line R. R. The Clyde S. S. Co. has a terminal there.

Bain's Marine Annual, 1926

Edited by Peter Bain, Editor of Shipping Illustrated, and Durward H. Primrose, Editor of The Marine Journal. 7¼ in. x 10 in. 314 pages. Price $2.50 net. Published by Merchant Marine Publishing Company, New York.

Intended to afford marine and allied interests a means to refresh their memories on outstanding personalities and events of the near and more distant past, Bain's Marine Annual which makes its first appearance this year is distinctive in character. Its sub-title, "Maritime Miscellany," indicates its character and scope. It is a collection of articles and notes, largely from the pens of the authors, published during the previous year, with extracts of recent government publications and a few special articles, a true miscellany. In respect to motorships and oil engines, the Annual contains few references, and such little credit as is given to them is of a grudging nature, as if the authors regret their coming and deplore the smashing inroads they have made on steam. Inasmuch as the annual makes no pretension to cover engineering development or ship operation, this blemish may be excused. The utility of the volume is that it is a compendium of general information, and the authors deserve to make a success of it.

Brown's Flags and Funnels

By F. J. N. Wedge. 5¼ in. x 8¼ in. 33 color plates and index. Published by James Brown & Son., Ltd., Glasgow.

A total of 660 house flags and stack marks belonging to shipping companies throughout the world has been assembled in a book of handy size by the well-known Scotch firm of nautical and marine engineering publishers. About 30 of the leading American steamship companies are represented in these pages. In the arrangement of the book there are 20 flags and stacks on each color plate, and each flag is numbered to facilitate reference from the index. To assist in identifying stacks and flags, a definite order is followed, the black stacks in the lead, followed by buff stacks, red stacks, and stacks in fancy colors, each group of colors showing first the plain stacks and then the stacks with distinguishing devices, these latter in turn being grouped, according to color, with few exceptions. In each case the house flag is alongside. The plates are printed by the offset process and are superior to the general run of flag reproductions. The principal variation from true colors occurs in the many shades of buff, for all of which a process yellow is used. The book deserves a place in every chart room.

Personal.

Rear-Admiral Chas. W. Dyson, U. S. N. (retired) has become associated with J. Barraja-Frauenfelder & Co. in the capacity of senior advisory engineer. All business matters of that firm pertaining to the powering of ships, including propeller designs, general power installations, investigations involving power problems, etc., will be subject to Admiral Dyson's final approval.

Charles P. Wetherbee has been retained by the Westinghouse Electric & Mfg. Co. as consulting engineer on marine work, in which capacity he will be attached to the South Philadelphia Works, of that firm. Mr. Wetherbee earned international fame during his 30 years of service with the Bath Iron Works, Ltd., Bath, Me., where he began his long career as a draftsman and rose ultimately to vice-president and superintending engineer.

Born in Detroit, Mich., in 1871, Mr. Wetherbee, after finishing his early school education, entered the Massachusetts Institute of Technology in 1887 and graduated in 1891. The same year he was appointed instructor in naval architecture, serving in that capacity until the following year, when he entered the Ecole d' Application du Génie Maritime, Paris, from which he graduated in 1894.

About that time a brilliant group of young American naval officers were polishing off their education abroad. Beuret, Woodward, Hasbrouck, Stocker and Snow were in Paris. Ferguson, Watt and Zahn were in Glasgow. Smith and Gilmor were in Greenwich, England. All these men have risen high in their profession, and it was with such a group that Mr. Wetherbee developed.

Shortly thereafter, Mr. Wetherbee was employed by the Newport News Shipbuilding & Dry Dock Co. in the office of the superintending naval constructor and remained there until 1895, when he obtained a position as draftsman at the Columbian Iron Works, Baltimore, whence he went to the Bath Iron Works in 1896.

In the vessels built at the Bath yard, naval craft predominated and among those for the machinery of which Mr. Wetherbee was responsible, were the battleship GEORGIA, the scout cruiser CHESTER and 25 destroyers.

Catalogs.

Bonney Wrenches, a 40-page miniature catalog of chrome vanadium and carbon steel wrenches. Bonney Forge & Tool Works, Allentown, Pa.

Pynolag Pyrometer Protection Tubes, a 2-page circular covering tubes for thermocouples. Louis C. Eitzen Co., 280 Broadway, New York City.

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